

REPORT 72-8846

FRACTIONAL WATT VUILLEUMIER CRYOGENIC REFRIGERATOR

W. S. Miller

V. L. Potter

The Garrett Corporation

AiResearch Manufacturing Co.

2525 West 190th Street

Torrance, California

March 1973

Final Report for Task I Preliminary Design

(NASA-CR-156770) FRACTIONAL WATT
VUILLEUMIER CRYOGENIC REFRIGERATOR Final
Report (AiResearch Mfg. Co., Torrance,
Calif.) 205 p

N78-77777

**Unclass
26428**

REPRODUCED BY
**NATIONAL TECHNICAL
INFORMATION SERVICE**
U.S. DEPARTMENT OF COMMERCE
SPRINGFIELD, VA. 22161

00/31

Prepared for

GODDARD SPACE FLIGHT CENTER

Greenbelt, Maryland 20771

209

REPORT 72-8846

**FRACTIONAL WATT
VUILLEUMIER CRYOGENIC REFRIGERATOR**

W. S. Miller

V. L. Potter

The Garrett Corporation

AiResearch Manufacturing Co.

2525 West 190th Street

Torrance, California

March 1973

Final Report for Task I Preliminary Design

Prepared for

GODDARD SPACE FLIGHT CENTER

Greenbelt, Maryland 20771

PREFACE

GENERAL

The Fractional Watt Vuilleumier Cryogenic Refrigerator Program was administered by the Goddard Space Flight Center, Greenbelt, Maryland, under Contract No. NAS 5-21715. The NASA Technical Monitor for this contract was Mr. Max Gasser.

The Task I preliminary design activities were completed by AiResearch Manufacturing Company, a division of The Garrett Corporation, in Torrance, California. The Program Manager was V. L. Potter and the Program Engineer was W. S. Miller. Major contributors to this Task I effort were V. L. Potter, mechanical design; M. Morimoto, stress analysis; M. Soliman, heat pipes; J. Denk, motor; and W. S. Miller, thermodynamics and heat transfer. In addition, C. W. Browning served as technical consultant.

OBJECTIVE

An engineering model long life VM refrigerator has been fabricated and delivered to NASA/GSFC under contract NAS 5-21096. The emphasis in the design of the engineering model was the attainment of long life. A natural extension to that work is application of the technology developed to flight type hardware. Thus, the objective of this program is to design and build a flight prototype VM refrigerator.

SCOPE OF WORK

The VM cryogenic refrigerator program consists of five major tasks: (1) Task I, Preliminary Design; (2) Task II, Analytical and Test Programs; (3) Task III, Model Specification; (4) Task IV, Model Fabrication and Test, and (5) Task V, Program Management and Periodic Documentation.

This technical report describes the Task I (Preliminary Design) phase of the VM cryogenic refrigerator design, development, and fabrication program. The general requirement of the refrigerator is to produce 0.25 watts of cooling at 65°K after operating 2 to 5 years in a space environment. These Task I accomplishments provide the basis for the further design, analysis, development testing, and evaluation to be conducted during the next phase (Task II) of this program.



CONTENTS

<u>Section</u>	<u>Page</u>
1 INTRODUCTION	1-1
Task 1 Preliminary Design	1-2
Subtask 1-1--Preliminary Engine Sizing and Thermal Analysis	1-2
Subtask 1-2--Preliminary Mechanical Design and Stress Analysis	1-3
Subtask 1-3--Heat Pipe Interface	1-3
Subtask 1-4--Motor Design and Specification	1-4
Subtask 1-5--Layout Drawings	1-4
Subtask 1-6--Summary Report	1-4
2 PRELIMINARY DESIGN SUMMARY	2-1
Introduction	2-1
Physical Description	2-1
System Operation	2-3
Performance and Design Summary	2-4
3 TECHNICAL DISCUSSION	3-1
Introduction	3-1
Design Requirements	3-1
Design Approach	3-1
Temperatures	3-4
Weight	3-4
Power	3-6
Pressure	3-6
Cycle Speed	3-6
Mechanical Assembly	3-6
Thermal Design	3-7
Introduction	3-7
Thermal Design Summary	3-8
Cold Regenerator Design	3-11



CONTENTS (Continued)

<u>Section</u>	<u>Page</u>
Hot Regenerator Design	3-29
Cold End Heat Exchanger Design	3-44
Hot End Heat Exchanger Design	3-45
Ambient Sump Heat Exchanger	3-49
Cold End Insulation	3-58
Hot End Insulation	3-66
Pressure Drop and Void Volume Tradeoff Factors	3-70
Flow Distribution Devices	3-76
Design of Cold End Seal	3-82
Design of Hot End Seal	3-88
Heat Pipe Interface	3-91
Design Requirements	3-94
Sink Temperature Selection	3-94
Heat Pipe Design	3-96
Heat Pipe Assembly Design	3-114
Interface Temperature Drop	3-116
Mechanical Design	3-117
Design Description	3-117
Cold End Closure Dome	3-149
Final Closure Technique	3-149
Stress Analysis	3-164
Introduction	3-164
Design Criteria	3-164
Discussion of Results	3-165
Motor Design	3-167
Appendix A REGNERATOR ANALYSIS	A-1
Appendix B FINAL CLOSURE SEALING WELDING TEST PLAN	B-1



ILLUSTRATIONS

<u>Figure</u>		<u>Page</u>
2-1	GSFC Flight Prototype Fractional Watt Vuilleumier Cryogenic Refrigerator Preliminary Design	2-2
3-1	Ultimate Strength of Inconel 718	3-5
3-2	Data Output From Idealized Cycle Program	3-9
3-3	Temperature Variation at Cold End of First Iteration Cold Regenerator	3-14
3-4	Pressure Drop of First Iteration Cold Regenerator	3-15
3-5	Temperature Variation at Cold End of Second Iteration Cold Regenerator	3-17
3-6	Pressure Drop of Second Iteration Cold Regenerator	3-18
3-7	Temperature Variation at Cold End of Preliminary Design Cold Regenerator	3-26
3-8	Pressure Drop of Preliminary Design Cold Regenerator	3-27
3-9	Cold End Flow Rate of Preliminary Design Cold Regenerator	3-28
3-10	Preliminary Design Cold Regenerator Thermal Loss per Cycle	3-30
3-11	Temperature Variation at Hot End of Preliminary Design Hot Regenerator	3-40
3-12	Pressure Drop of Preliminary Design Hot Regenerator	3-41
3-13	Hot End Flowrate of Preliminary Design Hot Regenerator	3-42
3-14	Preliminary Design Hot Regenerator Thermal Loss per Cycle	3-43
3-15	Schematic of Cold End Heat Exchanger	3-46
3-16	Schematic of Hot End Heat Exchanger	3-48
3-17	Sump Heat Exchanger Configuration	3-51
3-18	Rectangular Offset Plate-Fin-VM Refrigerator Sump Heat Exchanger	3-53
3-19	Colburn j Factor vs Reynolds Number for Sump Heat Exchanger Heat Transfer Surface	3-55
3-20	Fanning Friction Factor vs Reynolds Number for Sump Heat Exchanger Heat Transfer Surface	3-56
3-21	Cold End Temperature Profile for Heat Leak Calculations	3-59



ILLUSTRATIONS (Continued)

<u>Figure</u>		<u>Page</u>
3-22	Fractional Watt VM Cold Finger Thermal Analyzer Model	3-62
3-23	Temperature Distribution, Cold Finger of Fractional Watt VM	3-64
3-24	Hot End Temperature Profile for Radiation Heat Leak Calculations	3-69
3-25	Effect of Cold Void Volume on Ideal Refrigeration Tradeoff Values	3-74
3-26	Effect of Sump Void Volume on Ideal Refrigeration Tradeoff Values	3-75
3-27	Refrigeration Tradeoff Values as a Function of Void Volume Temperature	3-77
3-28	Typical Flow Distributor	3-78
3-29	Cold Displacer Sealing Design Configuration	3-83
3-30	Cold-End Seal Leakage--Pressure Drop Characteristics	3-85
3-31	Leakage Thermal Loss Model	3-86
3-32	Cold End Thermal Losses Due to Leakage as a Function of Leakage Rate	3-89
3-33	Hot-End Seal Configuration	3-90
3-34	Hot-End Seal Leakage Rate vs Pressure Drop	3-92
3-35	Hot-End Thermal Loss vs Leakage Rate	3-93
3-36	Space Radiator Design Chart	3-97
3-37	Effect of Radiator Area on Radiator Temperature	3-98
3-38	Transport Factors of Various Heat Pipe Working Fluids	3-99
3-39	Thermal Conductivity of Candidate Fluids	3-100
3-40	Pressure-Temperature at Saturation Conditions for Candidate Fluids	3-101
3-41	Candidate Wick Designs	3-104
3-42	Effect of Heat Pipe Length on Power Transport Capability	3-108
3-43	Effect of Temperature Level on Heat Pipe Performance	3-110
3-44	Effect of Temperature Level on Heat Pipe Performance	3-111



ILLUSTRATIONS (Continued)

<u>Figure</u>		<u>Page</u>
3-45	Schematic of the Test System	3-112
3-46	Overall Temperature Drop vs Power Transported	3-113
3-47	Aluminum Block Design Details	3-115
3-48	Proposed Fractional Watt VM Development Test Instrumentation	3-123
3-49	Break in Magnetic Coupling	3-125
3-50	DCA-VM Radial Interface - Concept A	3-127
3-51	DCA-VM Radial Interface - Concept B	3-128
3-52	DCA-VM Radial Interface - Concept C	3-129
3-53	DCA-VM Radial Interface - Concept D	3-130
3-54	DCA-VM Radial Interface - Concept E	3-131
3-55	DCA-VM Axial Interface Concept	3-133
3-56	HRC Proposed Radiometer-VM Interface	3-135
3-57	Flange Support	3-136
3-58	Tubular Frame Support	3-137
3-59	Cylindrical Shell Support	3-138
3-60	Hot End Mounting Configuration	3-139
3-61	Instrumentation Setup	3-141
3-62	Vuilleumier Engine Proximity Sensor System Calibration Test Setup	3-142
3-63	X-Y Plotter Curve for Probe S/N 41H450	3-143
3-64	X-Y Plotter Curve for Probe S/N 41X312	3-144
3-65	Proximity Probe Mounting Frame	3-145
3-66	Expanded Curve for Probe (SN 41H450)	3-146
3-67	Expanded Curve for Probe (SN 41X312)	3-147
3-68	Oscilloscope Traces	3-148
3-69	Cold End Vacuum Dewar Test Closure	3-150
3-70	Formed Channel	3-152
3-71	Riveted Assembly	3-152
3-72	Square Wire Retaining Ring	3-153
3-73	Split Ring Assembled With Bolts	3-154



ILLUSTRATIONS (Continued)

<u>Figure</u>		<u>Page</u>
3-74	Removable Split Ring Fixture--Hot Displacer Housing Connection to Sump	3-154
3-75	Removable Split Ring Fixture--Cold Displacer Vessel Connection to Sump	3-155
3-76	Bayonet Type Lock-Hot Displacer Housing Connection to Sump	3-155
3-77	Removable Split Ring Fixture Motor Housing Connection to Hot Displacer Housing	3-156
3-78	Bayonet Type Lock--Crankshaft Closure Dome Connection to Housing	3-156
3-79	Bayonet Type Lock--Motor Housing Connection to Hot Displacer Housing	3-157
3-80	Bayonet Type Lock--Cold Displacer Vessel Connection to Sump	3-157
3-81	Removable Split Ring Fixture Cold Displacer Vessel Connection to Sump	3-158
3-82	Cold End Closure Concept	3-159
3-83	Hot End and Crankshaft Closure Concept	3-160
3-84	Sump Preliminary Design	3-161
3-85	Cold End Preliminary Design	3-162
3-86	Hot End Preliminary Design	3-163
A-1	Heat Transfer and Pressure Drop Data for Stacked Screens	A-7
A-2	Gas Flow through an Infinite Randomly Stacked Sphere Matrix	A-8
A-3	Monel Specific Heat	A-10
A-4	Stainless Steel and Inconel Specific Heats	A-11
B-1	Candidate Closure Design	B-2
B-2	Restraining Tools - Typical	B-3
B-3	Hot-End Closure Assembly	B-4
B-4	Cold-End Closure Assembly	B-5
B-5	System Test Setup Diagram	B-7



ILLUSTRATIONS (Continued)

<u>Tables</u>		<u>Page</u>
2-1	GSFC Flight Prototype Fractional Watt Vuilleumier Cryogenic Refrigerator Summary of Nominal Performance and Design Values	2-5
3-1	Design Requirements--Flight Prototype--Fractional Watt Vuilleumier Cryogenic Refrigerator	3-2
3-2	Delta Sinusoidal Vibration Schedule--Sweep Applied Frequency Once Through Each Range	3-3
3-3	Delta Random Vibration--Applied in all Three Axes	3-3
3-4	Ideal Vuilleumier Cycle Analysis--Nomenclature Key	3-10
3-5	Hot End Power Requirements	3-11
3-6	Summary of Cold End Losses	3-12
3-7	Preliminary Design--Cold Regenerator--Performance Characteristics	3-19
3-8	Preliminary Design--Hot Regenerator--Performance Characteristics	3-33
3-9	Sump Heat Exchanger Design and Performance Summary	3-57
3-10	Thermal Analyzer Solution of Cold End Insulation with Aluminum Heat Shields	3-63
3-11	Cold End Insulation Heat Leak with Aluminum Heat Shields	3-65
3-12	Thermal Analyzer Solution of Cold-End Insulation Network with Rhodium-Plated Heat Shields	3-67
3-13	Heat Leak of Fractional Watt VM Cold Finger with Rhodium Plated Heat Shields	3-68
3-14	Input for K0270	3-71
3-15	Void Volume Tradeoff Parameters for Preliminary Design Fractional Watt VM Refrigerator	3-76
3-16	Fractional Watt VM Development Test Instrumentation	3-122
A-1	U.S. Sieve Series and Tyler Equivalents	A-9



SECTION I

INTRODUCTION

The progressing development of meteorological and communication spacecraft requires increasingly sophisticated sensing devices. A coexisting requirement is the development of cryogenic refrigeration; the sensitivity and performance of certain advanced sensing devices is enhanced when cooled to cryogenic temperatures, while others must be cooled to cryogenic temperatures in order to function.

Most NASA space applications for cryogenic refrigeration require operational lifetimes of 1 to 5 years. State-of-the-art production VM refrigerators, however, have life expectancies far less than the minimum acceptable for space applications. For this reason, NASA undertook the development of a long life cryogenic refrigeration system compatible with spacecraft requirements. The heat driven Vuilleumier (VM) cycle was the refrigeration system selected for development because of the following characteristics that are potentially advantageous for space applications: (1) long operating life due to low working stresses placed on bearings and internal seals, (2) use of thermal energy as the source of energy to drive the system, (3) small size and weight, (4) moderate efficiencies for low cooling capacity systems, and (5) low noise and vibration levels.

Government funded efforts in this area have shown significant progress in demonstrating the characteristics listed above, including long life. An engineering model long life VM refrigerator has been fabricated and delivered to NASA/GSFC under Contract NAS 5-21096. To demonstrate the feasibility of long life VM refrigerators for NASA space applications, Contract NAS 5-21715 was entered into in March 1972 between AiResearch and the NASA Goddard Space Flight Center. The contract covers design, development and fabrication of a flight prototype fractional watt VM refrigerator for integration into a radiometer device. The primary design requirements of the unit being developed are long operating life, with a 2 year operating life specified as a minimum requirement, and the use of design techniques and assembly methods compatible with unmanned satellite environmental conditions. The refrigerator shall be capable of cooling a 1/4 watt thermal load to 65°K.

The program under Contract No. NAS 5-21715 consists of the following major tasks:

- Task I Preliminary Design
- Task II Analytical and Test Program
- Task III Model Specification
- Task IV Model Fabrication and Test
- Task V Program Management and Periodic Documentation



The technical specification for this program is given by NASA/GSFC Statement of Work and Specification APD-763-28, dated September 15, 1971.

This report describes the Task I effort, which consisted of definition of the Preliminary Design of the Flight Prototype Fractional Watt Vuilleumier Cryogenic Refrigerator.

TASK I PRELIMINARY DESIGN

The principal goal of the Task I effort was to establish and recommend a preliminary design VM refrigerator which would meet the operational requirements for the intended application. The general design requirements are that the refrigerator cool a 1/4 watt load at 65°K. A summary of the preliminary design VM refrigerator characteristics is presented in Section 2 of this report. The detailed requirements and the technical discussion of the procedures utilized to arrive at the preliminary design configuration are presented in Section 3.

The Task I effort was delineated into several major subtasks, which are described briefly in the following paragraphs. The experience gained in the design and fabrication of the engineering model, GSFC 5 watt VM refrigerator, was utilized to the greatest possible extent during the present program.

Subtask I-1--Preliminary Engine Sizing and Thermal Analysis

The sizing of the refrigerator and its thermal analysis paralleled the mechanical design, which was a concurrent effort. The thermal analysis forms a major part of the design iteration loop, and thus was integrated into the overall preliminary design effort.

The initial design parameters used as the starting point for the thermal analysis were those derived for the conceptual design evolved at the beginning of the program. After establishing revised design parameters, including inputs from the mechanical design analysis and other associated tasks, the thermodynamic cycle was reevaluated. By repeating this process while holding the basic operating parameters constant--cycle pressure, hot-end temperature, ambient temperature, refrigeration temperature, and machine speed--the optimum physical characteristics of a refrigerator (based on the initial design parameters) were defined. Then, changing those basic parameters which are subject to variation for purposes of optimization, the characteristics based on a new set of parameters were defined. By comparing these successive iterations, the optimum preliminary design was selected.

Definition of the operating cycle described above allows determination of the required thermal characteristics of the various critical components of the machine. As these devices were analytically defined, modifications to the operating characteristics were dictated. These changes were fed back into the cycle analysis and the resulting changes in thermal requirements determined. The individual components were then modified, and the iterative procedure repeated. The entire VM refrigerator was thus iterated to its preliminary design, or present, configuration.



Subtask I-2--Preliminary Mechanical Design and Stress Analysis

The mechanical design considerations associated with VM refrigerator design are many and varied. Special attention was given to areas involving the drive mechanism, bearings, seals, refrigerant leakage, and contamination. Each of these were considered independently, as well as their interaction on one another and their impact on the thermal design.

The basic mechanical design approach developed for the previous engineering model machine was utilized for the fractional watt VM refrigerator. This approach makes use of solid compact bearings and non-contact seals to achieve long-life capability. Based on test results on the GSFC 5 watt VM, there is every reason to believe that the fractional watt VM refrigerator will meet all design requirements.

Two areas that required careful consideration in the mechanical design of the fractional watt machine are: (1) scaling down the drive mechanism and bearing approach used in the 5-watt machine to reduce weight and size and (2) the devising of a method of closure of the working volume of the engine that allows take-down of the engine for inspection during the run-in period and final close-out of the system by welding to prevent external leakage upon final assembly.

The preliminary mechanical design and stress analysis task also covered the structural design of the pressure containment members of the refrigerator and the dynamics of the system including the vibration loading of the key elements of the design. These analyses will be refined in Task 2.

Subtask I-3--Heat Pipe Interface

A preliminary heat pipe design was evolved to evaluate the removable heat pipe interface assembly. The heat pipe assembly design was only carried to the depth necessary to define the interface requirements. This, in turn, required definition of the number of heat pipes, their size, the working fluid, the wick configuration, and the materials of construction.

The heat pipe assembly design makes use of four separate heat pipes (two on each side of the refrigerator crankcase). This approach allows direct use of the heat pipe interface clamp design for the water-cooled interface. This offers a considerable advantage in that complete definition of the interface clamp requires extensive stress analysis to ensure good thermal contact under all operating conditions. With direct use of the identical clamp, duplicate effort was avoided.

It should be noted that the originally specified 180°F sump temperature was reduced to 140°F by program Technical Direction No. 1, thus allowing use of ammonia as the heat pipe working fluid.



Subtask I-4--Motor Design and Specification

A preliminary specification for the drive motor was prepared and submitted to the AiResearch motor group for review. It was determined that lamination dies were available in the size range and required pole pair number at AiResearch, the products of previous production fabrication orders. Based on this availability, further evaluation of other potential sources was eliminated.

The EMI specifications submitted by Honeywell for review were also subjected to analysis by the motor EMI Analysts. It is the consensus that the AC motor depicted in the VM layout drawing is a device that is free of radiated and conducted EMI. This is, however, an opinion since no one with the exception of Honeywell has tested at the levels required by their specification.

Subtask I-5--Layout Drawings

Layout drawings were developed as required in support of the above four Task I subtasks. A full system layout drawing has been constructed depicting the preliminary design. This drawing, together with the supporting analyses, form the basis for the Task I design review as well as the Task I Summary Report.

Subtask I-6--Summary Report

This Task I Summary Report describes the preliminary system design and summarizes the analyses conducted in Task I.



SECTION 2

PRELIMINARY DESIGN SUMMARY

INTRODUCTION

The GSFC fractional watt Vuilleumier cryogenic refrigerator is a gas-cycle, reciprocating machine, intended to operate continuously for five years at a design speed of 400 rpm. Continuous operation for such a long time period presents technical problems, namely long-life bearing and seal design. This technology has been developed, and is applied to a flight prototype machine for the first time in this design. As an aid in understanding the analysis described in subsequent sections of this Task I report, a brief physical and functional description and summary of the VM refrigerator is given in this section. The preliminary design layout drawing of this refrigerator is presented in Figure 2-1.

PHYSICAL DESCRIPTION

Although the primary energy input to the refrigerator is thermal energy, moving components (two displacers) are required so that the VM cycle will function properly. The two displacers (hot and cold) which make the refrigerator a reciprocating machine are driven by connecting rods attached to a crankshaft (Figure 2-1). Crankshaft throws are arranged 90-degrees apart so that the hot displacer is leading the cold displacer when the crankshaft is driven by an electric motor located at one end of the shaft. The displacers travel inside cylinders surrounded by packed-bed regenerators which in turn, are enclosed in pressure shells joined to the crankshaft housing, making the entire assembly pressure tight.

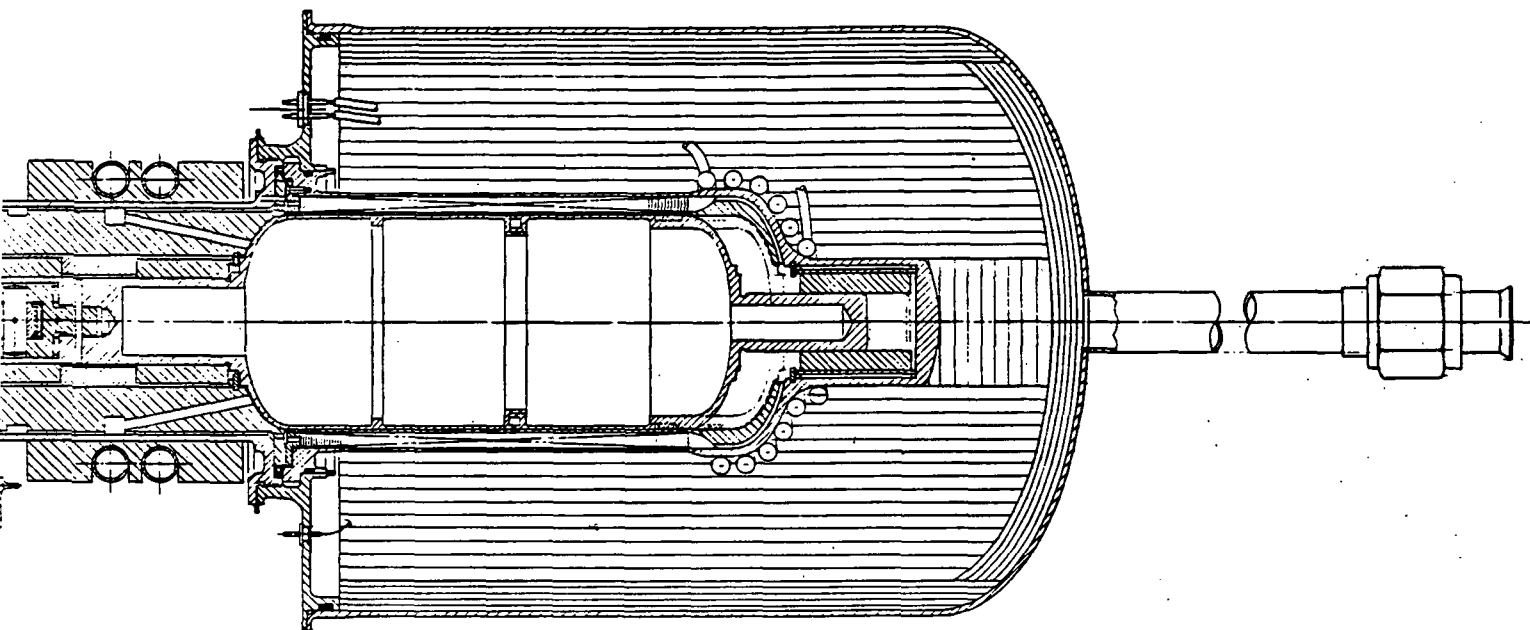
The displacers are arranged in a horizontally opposed configuration, with the cold displacer positioned at an angle of 180 degrees from the hot displacer, thus simplifying the crankcase and sump heat exchanger mechanical design and minimizing the interaction between the hottest and coldest parts of the system.

As indicated earlier, the primary energy for driving the refrigerator is introduced at the hot end in the form of heat. The interface at the hot end of the machine is shown in Figure 2-1. For this design, an electric heater is used to supply the heat; future spacecraft systems may use radioisotope heat sources or a solar collector to conserve solar cell produced energy.

Proper operation of the VM cycle also depends on rejecting heat in the crankcase-sump region. The GSFC VM refrigerator rejects heat to water cooling coils that interface with the crankcase and sump heat exchanger. These water cooling coils replace the ammonia heat pipes and a radiator which would be used in a flight-type system.

The refrigeration heat load is absorbed at the cold end via the cold-end heat exchanger. For test purposes, the refrigeration load is generated by a

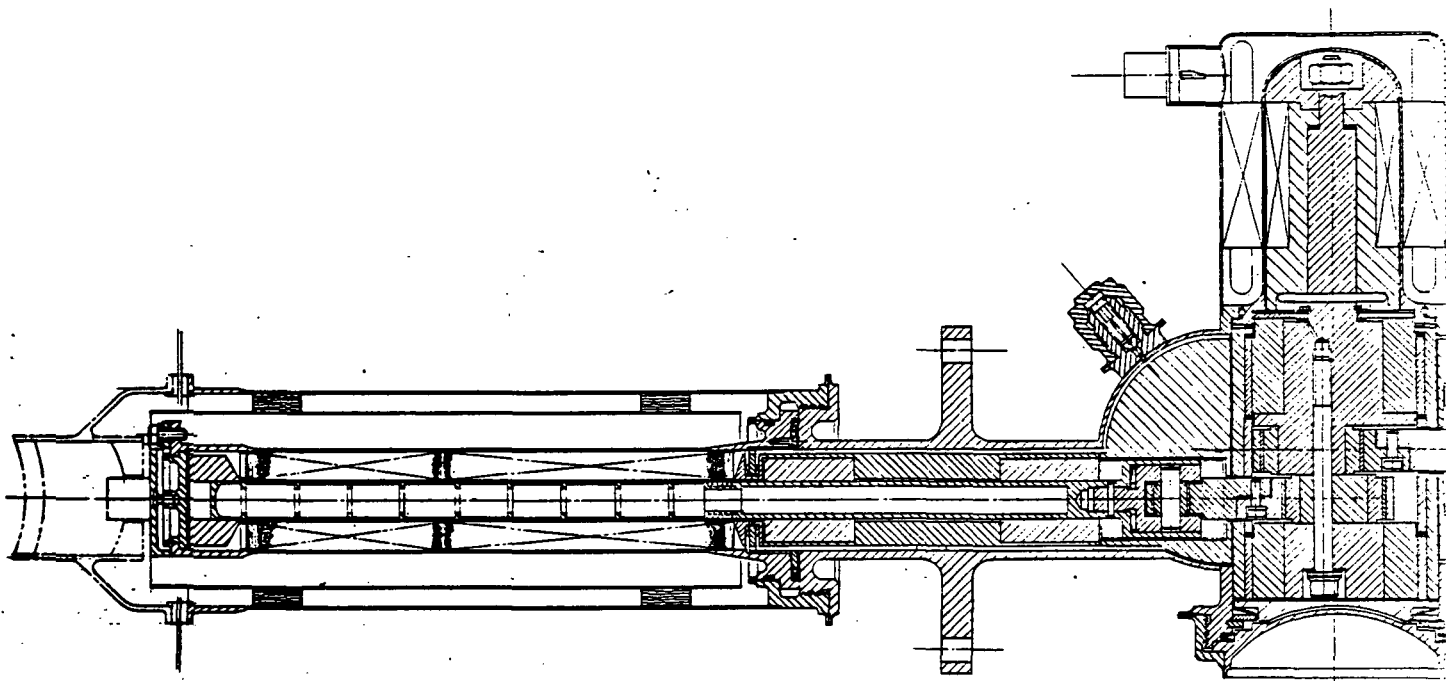




S-77604

Figure 2-1. GSFC Flight Prototype
Fractional Watt
Vuilleumier Cryogenic
Refrigerator Prelimi-
nary Design

22-a



2-2



AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California

small resistance-type heater bonded to the exterior surface of the cold-end heat exchanger. In a spacecraft system, the refrigerator load will be mounted directly to the cold end.

SYSTEM OPERATION

Externally, the VM refrigerator is viewed as a machine which absorbs heat at both a high and low temperature and then rejects this heat at some intermediate temperature (heat sink temperature). Heat input at the hot end of the machine provides the energy required to produce cooling at the cold end, while heat is being rejected from the sump or crankcase region of the system. The internal processes leading to the production of refrigeration are complex and difficult to describe in their real, rather than idealized form. Only a simplified description of system operation is provided here.

The total internal volume of the VM refrigerator is constant and consists of two parts: an active volume and a void volume. The active volume divides into three regions at different temperatures and of varying size due to reciprocating displacer motion. One of the active volumes is located at the hot end of the machine, one at the cold end, and the third in the crankcase-sump region. The remainder of the internal volume, or void volume, consists of fluid passages, component clearances and voids in the packed bed regenerators.

The arrangement of the two regenerators relative to the active volumes is such that gas passing from one volume to the next must first pass through a regenerator. In operation, gas passing through a regenerator is heated or cooled depending on flow direction. The regenerators are energy storage devices whose heat capacity greatly exceeds that of the gas. For example, gas flowing from the sump volume toward the hot volume is heated by the hot regenerator. The energy added to the gas was stored in the regenerator packing, or matrix, by gas flow in the reverse direction during a previous part of the cycle. The gas is heated to a temperature approaching that of the hot end by the time it exits the hot regenerator and enters the hot volume. The energy storage capability of the regenerator is generally such that its temperature profile is only slightly altered by providing energy to the gas. Flow in the reverse direction results in cooling of the gas by the regenerator until it enters the sump at a temperature approaching that of the heat sink.

The net effect of the operation of the regenerators can be viewed as isolating the three active volumes of the refrigerator in a thermal sense. Though this is not strictly true, the regenerators tend to maintain each active volume at nearly constant, but different temperature levels. These temperature levels are almost independent of the size of the volume, which is a function of crankshaft position. Thus with the hot displacer at the bottom of its stroke, the major part of the gas would be in the hot volume at an elevated temperature. With the hot displacer at the top of its stroke, the majority of the gas would be in the sump volume at a temperature approaching that of the heat sink. Remembering that the total internal volume is constant, as is the total mass of gas contained, rotation of the crankshaft brings about a change in the average temperature of the gas and an associated change in pressure. The pressure drop in the regenerators is very small and thus the pressure throughout the internal



volume of the machine is nearly the same at a given crankshaft position. This pressure depends on the amount of gas at a given temperature and therefore varies cyclically as the crankshaft is rotated. The only function of the drive motor is to provide the reciprocating motion of the displacers, not to provide mechanical work to the working fluid.

With the above processes described, the VM refrigerator can be viewed as a Stirling-type expansion machine (cold end) driven by a thermal compressor (hot end). Hot displaced volume is generally large compared to the cold displaced volume. Thus, the shuttling of gas between the hot and sump active volumes via the hot regenerator is the major factor causing the internal pressure to vary in a cyclic manner. These pressure variations (decreased slightly by pressure drop) are imposed on the cold end of the machine. By proper phasing of the cold displacer motion relative to the hot displacer, the cold displaced volume can be operated as an expansion volume to produce the cooling effect. A 90 degree phase angle, with the hot displacer leading the cold displacer, has been shown to provide near optimum performance.

The production of the cooling effect in the cold end of the machine can be viewed as follows: The gas entering the cold volume is cooled to near the refrigeration temperature by the cold regenerator. This gas then undergoes an expansion thereby further reducing its temperature and providing the capability to absorb heat from the refrigerator load. The expansion or reduction in pressure is brought about primarily by the processes taking place in the hot and sump regions of the machine rather than movement of the cold displacer. The phasing of the cold displacer movement is set to take maximum advantage of the pressure variations established within the unit. Subsequent to the expansion process, the gas is then returned to the regenerator at a temperature slightly below the original incoming temperature. This serves to reestablish the temperature profile in the regenerator matrix by absorption of heat from the packing and makes up for regenerator inefficiencies. The regenerator is thus placed in a thermal status to cool incoming gas during the next cycle process.

PERFORMANCE AND DESIGN SUMMARY

A summary of major performance and design parameters for the GSFC fractional watt Vuilleumier cryogenic refrigerator is given in Table 2-1.



GSFC FLIGHT PROTOTYPE FRACTIONAL WATT VUILLEUMIER CRYOGENIC REFRIGERATOR
SUMMARY OF NOMINAL PERFORMANCE AND DESIGN VALUES

72-8846
Page 2-5

SECTION 3

TECHNICAL DISCUSSION

INTRODUCTION

This section presents a detailed discussion of all of the analyses performed leading to definition of the recommended preliminary design fractional watt Vuilleumier refrigerator. The characteristics of the design have been summarized in Section 2 of this report. This section provides the reasoning and supporting data leading to the selection of the particular configuration of each major component of the machine.

The design requirements are presented in detail in order to completely define the overall objectives. Next the design approach is discussed, with particular emphasis on the basis of selection of each of the critical operating parameters and mechanical features of the machine. Under thermal design, both the individual components and development of optimization tradeoff parameters are presented. The heat pipe interface is discussed, including the heat pipes themselves, the interface clamp, and the sump water cooling coils. The long life aspects of the refrigerator, as well as stress analysis and interfacing requirement details, are presented under the subsection entitled mechanical design. Finally, the specialized features of the refrigerator drive motor are discussed in detail.

DESIGN REQUIREMENTS

The flight prototype fractional watt Vuilleumier cryogenic refrigerator has been designed to meet the requirements of NASA/GSFC Statement of Work APB-763-28, dated September 15, 1971. The detail requirements are summarized in Table 3-1. In addition to the detail requirements specified, the refrigerator is to interface with a radiometer device. Considerable attention has been paid to the interfacing requirements, both through close coordination with NASA and the radiometer manufacturer, and by detail design of various candidate interface component concepts. Thus not only the detail requirements have been considered, but also the intent of the program. This summary is presented in order to assure clear understanding of all program requirements.

DESIGN APPROACH

The general approach taken in the design of the VM refrigerator has been one of achieving all of the specified requirements. The power specified has dictated minimization of all losses. Considerable effort has been expended in developing methods of joining the various components of the refrigerator with a minimum of interface weight. The primary objectives, however, have been absolute assurance of the required refrigeration capacity and long life.



TABLE 3-1
DESIGN REQUIREMENTS
FLIGHT PROTOTYPE
FRACTIONAL WATT VUILLEUMIER CRYOGENIC REFRIGERATOR

Parameter	Requirement
Thermal	
Refrigeration Heat Load	0.25 watt
Refrigeration Temperature	65°K
Thermal Input	80 watts maximum
Heat Rejection Temperature	333°K maximum
Operating Life	Two years minimum at 100% duty cycle, unattended in a spacecraft environment
Mechanical	
Configuration	In-line
Weight, Including Motor	18 lb maximum
Heat Rejection Mechanism	Water cooled sump interface, compatible with heat pipes
Heat Supply Mechanism	Electrical heater conductively interfaced with the hot cylinder
Size	Not specified
Electric Motor Power	10 watts maximum
Environmental*	
Shock	Not specified
Humidity	90% $\pm$ 3% at 30°C $\pm$ 2°C for 24 hours
Vibration	3 axis, sinusoidal and random motion
(a) Sinusoidal-Sweep Frequency	See Table 3-2
(b) Random Motion	See Table 3-3
Prelaunch	24°C $\pm$ 2.8°C, 50% relative humidity maximum, for 3 weeks
Sand and Dust	Not specified
Design Load Factors	30 g in any direction
Storage Temperature	-17.8°C to +54.5°C up to 3 years
Storage Pressure	1 atm to 10 ⁻¹⁰ torr up to 3 years
Attitude	Operational or transportable in any attitude
Acceleration	Operable under 5 g perpendicular to inline axis
Temperature	Operable in a vacuum at 0 to 120°F steady state or cycling between 0 and 120°F with a minimum cyclic period of 6.2 hours.
Operating Pressure	Operable in ambient pressure from 1 atmosphere to 10 ⁻¹⁰ torr for five years

*Testing to confirm compliance with these requirements is not required.



TABLE 3-2

DELTA SINUSOIDAL VIBRATION SCHEDULE
SWEEP APPLIED FREQUENCY ONCE THROUGH EACH RANGE

THRUST AXIS Z-Z		
Frequency Range (Hz)	Level (0-to-peak)	Sweep Rate (Octaves/minute)
5-11	12.2 mm. d.a. (0.48 in. d.a.)	2.0
11-17	±2.3 g	2.0
17-23	±6.8 g	1.5
23-200	±2.3 g	2.0
LATERAL AXES X-X AND Y-Y		
5-9.9	19.0 mm d.a. (0.75 in. d.a.)	2.0
7.1-14	±2.0 g	2.0
14-200	±1.5 g	2.0

TABLE 3-3

DELTA RANDOM VIBRATION
APPLIED IN ALL THREE AXES

Frequency Range (Hz)	PSD level (g^2/Hz)	Acceleration (g - rms)	Duration
20-300	0.0029 to 0.045 increasing from 20 Hz at rate of 3dB/octave	9.1	4 minutes each axis
300-2000	0.045	9.1	4 minutes each axis



In the process of synthesizing the design of the refrigerator, many tradeoffs were made in arriving at the preliminary design recommended in this report. These tradeoffs were made based on quantitative evaluations as well as qualitative considerations. Much of the experience gained during the GSFC 5 watt refrigerator development program has been applied directly in the evolution of this design.

Details of each design area and its design selection are discussed in another portion of this section; however, an overall summary of major design considerations is presented in the following paragraphs.

Temperatures

The selection of the design temperatures was based on system requirements of the overall refrigerator system. The cold end cooling temperature of 65°K (117°R) was specified as a design requirement; by calculating the internal gas heat transfer coefficients, the average gas temperature for the cold expansion volume was found.

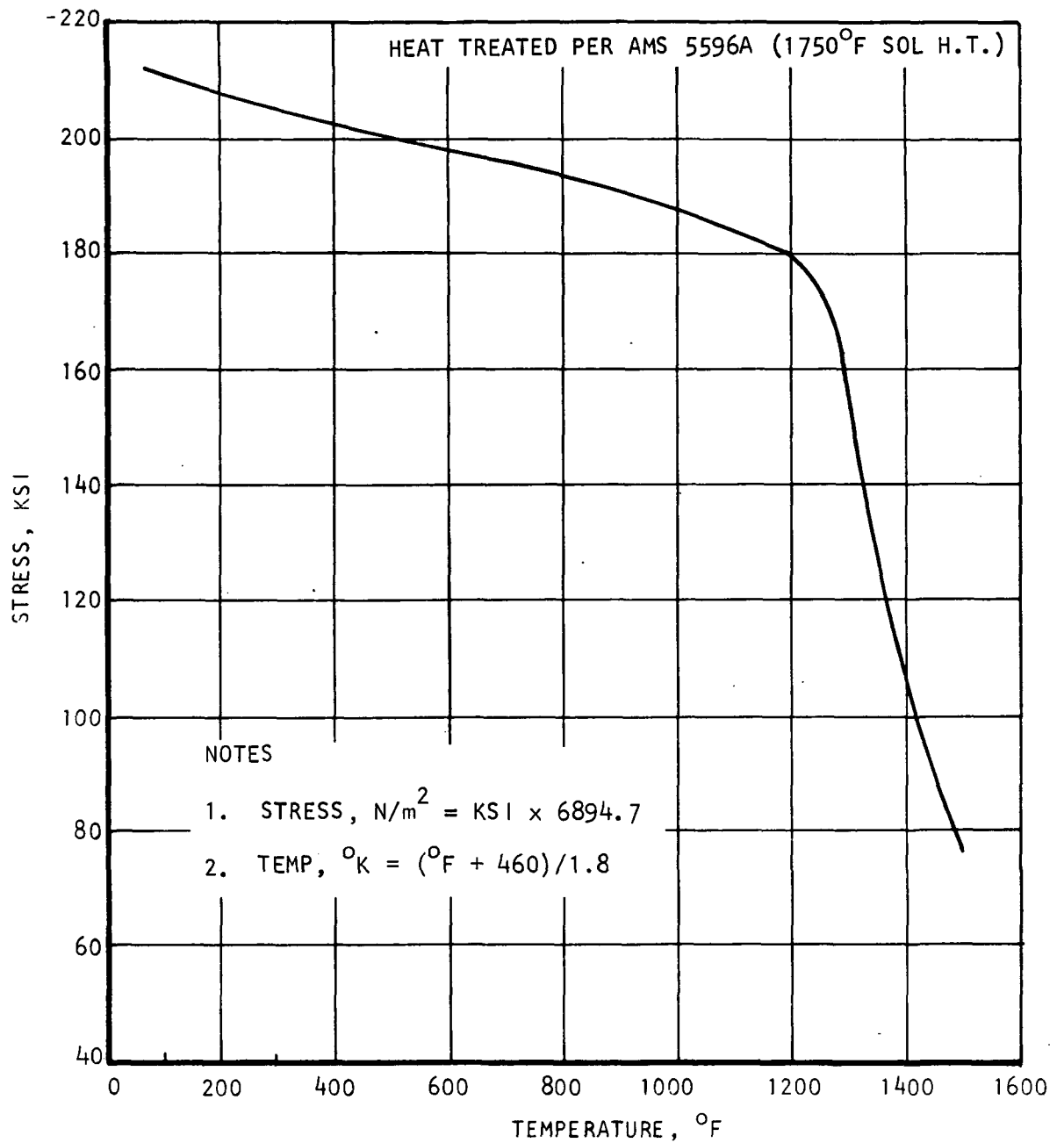
The gas temperature of the sump is primarily controlled by the heat rejection system. For this system, the maximum coolant temperature at the sump is specified as 333°K (600°R). An average sump gas temperature of 344°K (620°R) was selected by adding the temperature drops across the sump heat exchanger, the interface between the sump and the sump clamp, the clamp itself, the interface between the clamp and the water cooling coil, and the film temperature drop between the coil and the cooling water.

The selection of the hot end temperature was based primarily on material considerations. With the use of electrical heating, the power source temperature does not play an important part in the selection of the hot end temperature. Heaters are available which are capable of operation at temperatures considerably above the maximum operating temperature of the hot end pressure containment dome. From a thermodynamic standpoint, the higher the hot end temperature, the higher the thermal efficiency. The selected temperature of 867°K (1100°F) was based on the strength characteristics of Inconel 718 and also on the bearing characteristics of Boeing Compact 6-84-1. Figure 3-1 shows the sudden drop in strength of Inconel 718 at temperatures above 992°K (1200°F). A 55.6°K (100°F) design margin of safety was allowed below this value. The hot end is designed structurally for pressure containment of 992°K (1200°F); thus the use of 867°K (1100°F) hot end temperature represents a margin of safety on performance, and not stress level. Based on heat transfer analysis of the hot end heat exchanger, the average gas temperature in the hot expansion volume is 853°K (1535°R).

Weight

As previously indicated, the weight of the refrigerator was considered in detail in the design. Minimum weight materials have been utilized wherever possible. An example of this is the sacrifice of thermal performance incurred by the use of aluminum instead of copper for the sump interface clamp. The weight of the moving elements of the refrigerator has been minimized in order to reduce both the induced dynamic loads and total weight. The assembly concepts have been developed with minimum weight a primary consideration.





S-77576

Figure 3-1. Ultimate Strength of Inconel 718



Power

Power was also a major concern in the design of the refrigerator. In the process of synthesizing the design, considerable attention has been focused on losses. The thermal power input was minimized by requiring very low thermal losses. The motor power requirement has been kept low by maintaining a very low pressure drop across the hot displacer.

Pressure

The cycle pressure was selected in a somewhat qualitative manner. Thermodynamically, the higher the operating pressure, the higher the refrigeration capacity (or the smaller the size of the refrigerator). A maximum operating pressure of $10.33 \times 10^6 \text{ N/m}^2$ (1500 psia) was selected in trying to minimize the effect of axial conduction losses for the cold end. For low pressures, the wall thickness of the cold end is determined by fabrication limitations. However, as the pressure increases, the wall thickness requirement remains constant until pressure level, rather than fabrication requirements, becomes the design consideration. By thermally designing the refrigerator to operate at $6.895 \times 10^6 \text{ N/m}^2$ (1000 psia) and by structurally designing the refrigerator for $10.33 \times 10^6 \text{ N/m}^2$ (1500 psi), the pressure of the working gas can later be increased by 50 percent 6.895×10^6 (1000) to $10.33 \times 10^6 \text{ N/m}^2$ (1500 psia), if necessary, to increase the capacity of the engine. This provides a margin of almost 50 percent in the thermal refrigeration capacity. The maximum operating pressure was multiplied by 1.67 to determine the proof pressure, and by 2.25 to obtain the design burst pressure.

Cycle Speed

The speed of the refrigerator was selected based on the extensive investigations performed in support of the GSFC 5 watt VM refrigerator. At very low speeds, the residence times of the gas in the heat transfer devices are such that near isothermal operation can be achieved. This would, however, require a refrigerator of relatively large size, and the other thermal losses, such as axial conduction, would eliminate any gains in thermal performance. If, on the other hand, the refrigerator was operated at a very high speed, the temperature response of the gas and heat exchanger devices would be such that the thermal performance of the refrigerator would be greatly decreased.

A speed of 400 rpm was selected as a good compromise. This speed also results in a very low bearing speed and low dynamic loading for the bearings to support. To further increase life, it may be possible to reduce the speed, with an accompanying reduction in capacity.

Mechanical Assembly

The approach taken here has been the use of mechanical joints which allow the refrigerator to be disassembled after checkout tests. Upon verification of adequate performance, all joints may be permanently closed by welding without disturbing the mechanical integrity. Temporary non-organic seals are utilized at all assembly interfaces. Provisions have been incorporated for destruction



of these seals prior to final joint closure. In this manner, the attainment of absolute leak tight final closure welds is assured. Many mechanical closure concepts were investigated, with the emphasis on attainment of the goals described above with minimum weight.

THERMAL DESIGN

Introduction

The thermal design of a VM refrigerator is a lengthy iterative process. Initially, rough cut design calculations are made in order to establish the design approach and basic sizes of the machine's elements. After the basic design is defined, effort must be concentrated on matching the thermodynamic design with that of the heat transfer devices (heat exchangers and regenerators). Typically, the configurations and volumes of the heat transfer devices are adjusted to improve their heat transfer and pressure drop characteristics. These adjustments imply that changes be made to the active displaced volumes to compensate for the influence of the heat transfer devices on the thermodynamic processes of the working fluid. In turn, once the active volumes are changed, the heat transfer devices require adjustment to account for the variations in flows, pressure levels, and heat loads. This iterative process is carried out until the thermodynamic cycle parameters match the designs of the heat transfer devices. By examining several matched designs, a near-optimum refrigerator can be selected.

In this program, emphasis has not been placed on complete optimization of the refrigerator with respect to thermal performance; long operational life and reliability were considered the most important. It is believed, however, that due to the care taken in the detail thermal design, a near-optimum thermal design has resulted.

This Task I report presents the preliminary design of the fractional watt VM. As detail design progresses, modifications to the preliminary design components will be made as necessary. Where possible, the experience gained in the design and fabrication of the GSFC 5 watt VM refrigerator has been applied to the configurations presented herein. Based on this background, the design changes necessary as detail design progresses should be minimized. This section summarizes the design analyses; the topics covered are:

- Thermal Design Summary
- Cold-end regenerator
- Hot-end regenerator
- Cold-end heat exchanger
- Hot-end heat exchanger
- Ambient heat exchanger
- Cold-end insulation



- Hot-end insulation
- Pressure drop and dead volume tradeoff factors
- Cold-end flow distributor
- Design of cold end seal
- Design of hot end seal

The general method of analysis applied to both the hot and the cold regenerators, and the regenerator matrix characterization is presented in Appendix A of this Task I report.

Thermal Design Summary

The thermal design of the VM refrigerator is a lengthy iterative process; therefore, the cycle and thermal design are summarized before the detailed discussions. Many of the thermal features of the preliminary design VM have been presented in Section 2 of this report. This sub-section gives details of the thermodynamic cycle and the breakdown of the hot end power and cold end losses.

1. Thermodynamic Cycle

The thermodynamic cycle is analyzed by use of the ideal cycle analysis computer program. The information obtained includes the cyclic pressure profile; the mass flows in various portions of the machine, the ideal refrigeration, and the heat input required. Figure 3-2 presents the program output for the operating conditions of the preliminary design fractional watt VM refrigerator. The effects of pressure drop on the hot and cold-end have been accounted for in the calculations. Table 3-4 gives the nomenclature utilized in Figure 3-2.

It should be noted that the output from the ideal cycle analysis program forms a major portion of the input for the regenerator analysis. This analysis procedure is presented in Appendix A.

2. Hot End Power

The hot end power input is specified at 80 watts. This power input is divided into several portions: (1) the ideal required input, from Figure 3-2, (2) the losses to ambient, and (3) losses to the sump region. The losses have been calculated based on the preliminary design configuration of the VM refrigerator. The calculated total is slightly less than the maximum allowable, which provides a margin of safety for changes incurred as detail design progresses. The calculated hot end power distribution is presented in Table 3-5.



FRACTIONAL WATT VM-PRELIMINARY DESIGN UNIT WITH ESTIMATED DELTA P

OPERATING PARAMETERS

COLD VOLUME TEMP. ■ 112.00 R
 SUMP VOLUME TEMP. ■ 620.00 R
 HOT VOLUME TEMP. ■ 1535.00 R
 COLD REGEN. TEMP. ■ 366.00 R
 HOT REGEN. TEMP. ■ 1077.50 R
 COLD DISPLACED VOL. ■ .05530 CU-IN
 HOT DISPLACED VOL. ■ 1.88500 CU-IN
 COLD DEAD VOL. ■ .10000 CU-IN
 SUMP DEAD VOL. ■ 3.00000 CU-IN
 HOT DEAD VOL. ■ .35000 CU-IN
 COLD REGEN. VOL. ■ 1.32800 CU-IN
 HOT REGEN. VOL. ■ 3.33300 CU-IN
 GAS CONSTANT ■ 4634.40 IN-LB/LBM-R
 SPEED ■ 400.00 RPM

 CHARGE PRESSURE ■ 753.41 PSIA
 CHARGE TEMPERATURE ■ 535.00 R
 MASS OF FLUID ■ .0029 LBM
 TOTAL VOLUME ■ 10.05130 CU-IN

PRESSURE ANGLE DEG	MASS PRESS PSIA	FLOW VC CU-IN	PROFILE VA CU-IN	VH CU-IN	MDOTC LB/SEC	MDOTA LB/SEC	MDOTH LB/SEC	MDOTRCA LB/SEC	MDOTRHA LB/SEC
20.	969.93	.1017	3.6738	1.6148	.00102	-.00959	.00551	.00264	.00695
40.	986.05	.1065	3.3856	1.8982	.00154	-.00848	.00462	.00277	.00571
60.	996.63	.1138	3.1678	2.1086	.00186	-.00614	.00304	.00252	.00362
80.	999.98	.1228	3.0468	2.2206	.00193	-.00286	.00099	.00190	.00096
100.	995.56	.1324	3.0371	2.2207	.00172	.00082	-.00120	.00101	-.00183
120.	984.08	.1415	3.1399	2.1089	.00128	.00430	-.00317	-.00000	-.00430
140.	967.36	.1488	3.3428	1.8987	.00088	.00708	-.00463	-.00097	-.00609
160.	947.82	.1536	3.6214	1.6153	.00003	.00877	-.00542	-.00176	-.00701
180.	928.05	.1553	3.9419	1.2931	-.00059	.00932	-.00552	-.00230	-.00703
200.	910.36	.1536	4.2659	.9707	-.00110	.00879	-.00498	-.00254	-.00625
220.	896.58	.1488	4.5543	.6872	-.00147	.00734	-.00394	-.00250	-.00484
240.	888.01	.1415	4.7722	.4766	-.00168	.00519	-.00253	-.00220	-.00299
260.	885.35	.1325	4.8934	.3645	-.00171	.00255	-.00089	-.00169	-.00087
280.	888.84	.1229	4.9033	.3642	-.00157	-.00034	.00084	-.00100	.00134
300.	898.18	.1138	4.8006	.4758	-.00127	-.00326	.00252	-.00020	.00346
320.	912.58	.1065	4.5979	.6859	-.00081	-.00594	.00399	.00065	.00529
340.	930.67	.1017	4.3195	.9692	-.00024	-.00810	.00509	.00148	.00662
360.	950.56	.1000	3.9989	1.2914	.00039	-.00941	.00565	.00218	.00723

THE CYCLE PRESSURE ABOVE IS IDEAL, THE P-V INTEGRALS HAVE BEEN MODIFIED FOR PRESSURE DROP
 HOT MAXIMUM DP= 1.0000 PSI COLD MAXIMUM DP= 2.5000 PSI

IDEAL REFRIGERATION AND HEAT INPUT

REFRIGERATION ■ 3.5513 WATTS
 THERMAL HEAT ■ 27.0030 WATTS
 MAX. PRESSURE ■ 1000.0000 PSIA AT ANGLE ■ 78.59 DEGREES

Figure 3-2. Data Output From Idealized Cycle Program



TABLE 3-4
IDEAL VUILLEUMIER CYCLE ANALYSIS
NOMENCLATURE KEY

Symbol	Definition
Press	Cycle Pressure
Angle	Crankshaft angle referenced to cold displacer top dead center
VC	Cold displaced volume
VA	Ambient displaced volume
VH	Hot displaced volume
MDOTC	Flow rate into cold volume
MDOTA	Flow rate into ambient volume
MDOTH	Flow rate into hot volume
MDOTRCA	Flow rate into cold regenerator at the end toward the sump
MDOTRHA	Flow rate into hot regenerator at end toward the sump



TABLE 3-5
HOT END POWER REQUIREMENTS

	Heat Input, Watts
Regenerator Loss ($\dot{M} c_p \Delta T$)	5.67
Regenerator Inner Wall Conduction Loss	6.34
Regenerator Bed Conduction Loss	3.71
Regenerator Outer Wall Conduction Loss	12.82
Displacer Conduction Loss	15.33
Insulation Loss	5.90
Seal Leakage Loss	0.54
Ideal Heat Input (Includes ΔP Correction)	<u>27.00</u>
Total Calculated Heat Input	77.31 watts

3. Cold End Losses

The cold end losses are calculated in a similar manner to the hot end. In this case, however, the calculated losses represent heat introduced to the cold end in addition to the refrigeration heat load. Thus the sum of the losses and the refrigeration requirement represents the total heat that must be carried to the sump region and rejected to the heat sink. These losses are summarized in Table 3-6. It should be noted that the displacer gas leakage loss is based on very conservative assumptions and bearing clearances representative of two years of operation. The bearing wear rate and thus the clearances after two years are also considered conservative. Thus it is expected that the refrigerator will be capable of somewhat greater net cooling than indicated in Table 3-6 after two years of operation.

Cold Regenerator Design

1. Introduction

The cold regenerator is one of the most important components of the VM refrigerator, it not the most important. The regenerator cools gas as it flows from the sump region of the machine to the cold expansion volume. The gas expansion process further reduces the temperature, providing cooling at the cold temperature. After heat absorption from the refrigeration load, the gas is returned to the sump through the regenerator. This reverse flow process heats the gas by removal of energy stored in the regenerator matrix, and reestablishes the matrix temperature profile for cooling the gas during the next cycle. The periodic storage and removal of energy from the matrix allows the working fluid to pass from one essentially constant temperature portion of the machine to another. An ideal regenerator may thus be viewed as a temperature isolator.



TABLE 3-6
SUMMARY OF COLD END LOSSES

	Heat Loss, Watts
Regenerator Loss ($\int \dot{M} c_p dT$)	0.929
Regenerator Inner Wall Conduction Loss	0.200
Regenerator Bed Conduction Loss	0.654
Regenerator Outer Wall Conduction Loss	0.629
Displacer Conduction Loss	0.256
Insulation Loss	0.047
Displacer Leakage Loss (2 year worn bearings)	<u>0.569</u>
Total losses	3.284
Ideal Refrigeration (Includes ΔP Correction)	3.551
Ideal Q Minus Losses = Net Refrigeration (after 2 years operation)	0.267 (0.25 req'd)

The design requirements for an efficient regenerator demand that it must (1) absorb heat from the gas stream while at nearly the same temperature as the gas, (2) store this energy without significant temperature change in a given matrix locality, and (3) resupply the energy to the gas stream when the flow reverses direction, again while at a temperature very closely approaching that of the gas. These design requirements dictate that the regenerator packing (1) must have a very large heat capacity relative to that of the gas, (2) must have large values of heat transfer coefficient, heat transfer area, and thermal diffusivity, and (3) must be configured to limit axial conduction of heat from one end to the other. In addition, for the VM application, the void volume in the regenerator must be minimized. Obtaining these desirable regenerator characteristics from a specific packing generally leads to increased pressure drop as the thermal characteristics improve. Since pressure drop as well as void volume degrades the overall refrigerator performance, the basic tradeoff in the design of a regenerator is between heat transfer potential and the detrimental factors accompanying this potential.

2. Method of Analysis

The analysis of the regenerators for a VM refrigerator cannot be based on the classical effectiveness parameters that make use of end point temperatures. The system pressure fluctuates and a considerable amount of gas is stored in the regenerator void volume during various parts of the flow cycle. This periodic mass storage characteristic, coupled with the basic transient nature of regenerators, dictates that a finite difference technique be used in the analysis.



AiResearch has developed a computer program which utilizes finite difference techniques to analyze regenerators. The required program inputs are the regenerator physical characteristics, the matrix heat transfer and pressure drop characteristics, and the initial and boundary conditions. The computer program and the general characteristics of both spherical shot and screen regenerator packing materials are discussed in detail in Appendix A of this report.

The required boundary conditions are the time dependent pressures and flow rate profiles and the gas temperature at the sump end of the regenerator. The cold end gas temperature under reverse flow conditions is also required. These parameters are obtained from the design requirements of the VM, and the idealized cycle analysis computer program.

3. Initial Iterations

The regenerator design concept studies performed on the 5 watt VM program (Ref. 3-1) clearly indicated the advantages of the annular regenerator, as compared to those located inside the displacers or those located in parallel with the cold displacer. Therefore, the annular regenerator was the only type considered for the present machine.

The cold regenerator design was evolved along with the rest of the VM through numerous design iterations. However, certain initial ground rules were utilized which simplified the process. The cold displacer diameter was fixed early in the program, thus defining the inner diameter of the annulus. It was decided that, if at all possible, the packing would be composed entirely of spherical shot, in order to provide the maximum matrix heat capacity. The ground rule was also established that the annular height would be 30 shot diameters. This is necessary in order to prevent channeling of the gas along the regenerator wall.

The first regenerator analyzed utilized a packing material composed of 0.000178m (0.007 in.) diameter monel shot. The annular height was 0.00566m (0.21 in.), and the regenerator length was 0.1132m (4.5 in.). The gas and metal temperatures at the cold end are presented in Figure 3-3 as a function of crankshaft angular position. Figure 3-4 presents the pressure drop characteristic of this regenerator.

Figure 3-3 indicates that the heat transfer coefficient between the gas and the spherical shot matrix is sufficiently high. The maximum temperature deviation between the gas and the metal, except at the instant of flow reversal, is approximately 0.167°K (0.3°R). However, the heat capacity of the matrix is insufficient. This is evidenced by the gas temperature deviation of 1.333°K (2.4°R), which is considered excessive. The maximum pressure drop of $2.76 \times 10^4 \text{ N/m}^2$ (4.0 psi), as shown in Figure 3-4, is also considered too high.

The regenerator deficiencies pointed out above led to the next design iteration. The first (warmer) half of the regenerator was filled with 0.000254m (0.010 in.) diameter monel shot, and the annular height was increased to 0.00762m (0.3 in.). Both of these changes reduce pressure drop and increase matrix heat capacity. The second (colder) half of the regenerator was packed with 0.000178m (0.007 in.) diameter shot, in order to maintain the high heat transfer coefficient.



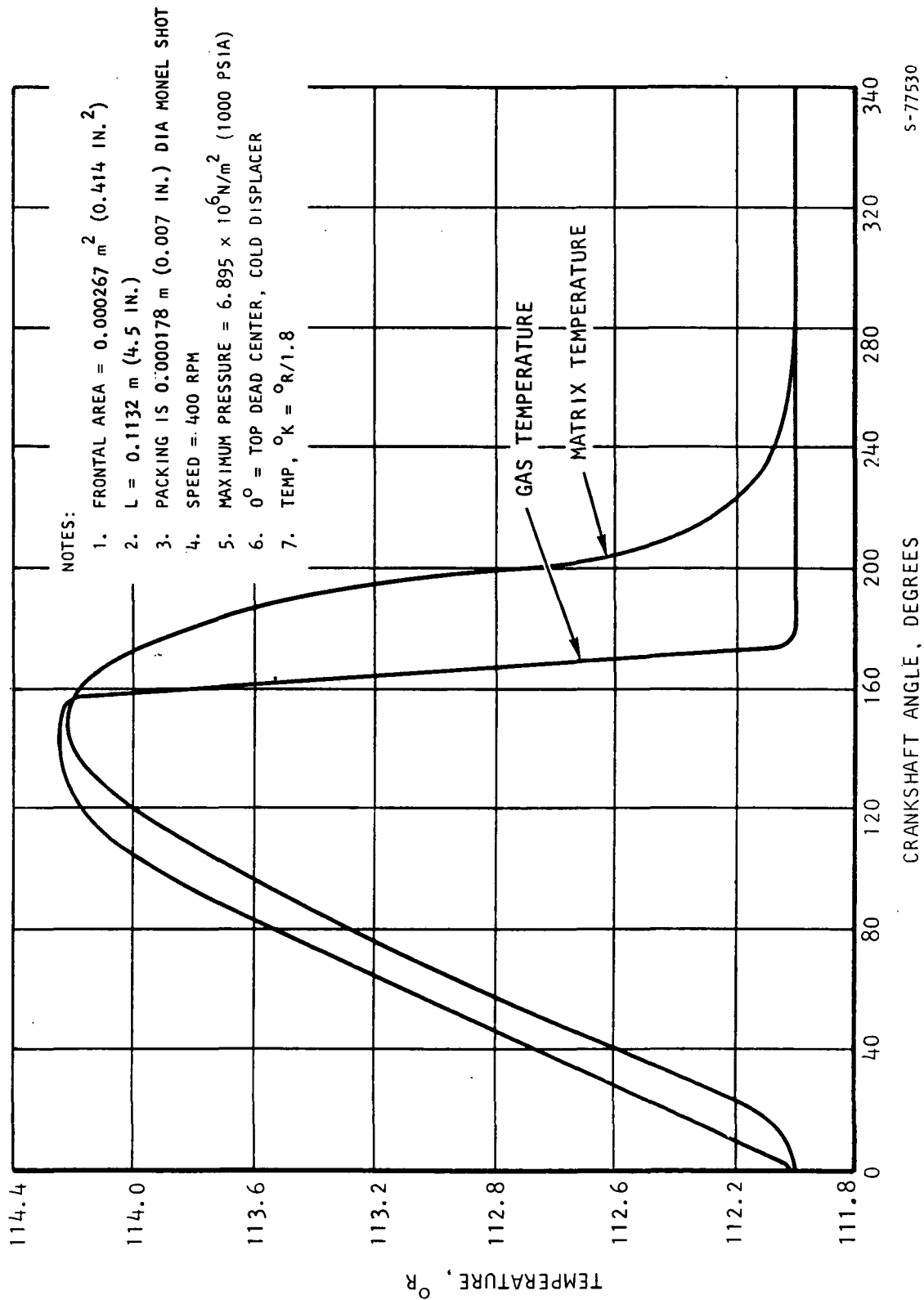


Figure 3-3. Temperature Variation at Cold End of First Iteration
Cold Regenerator

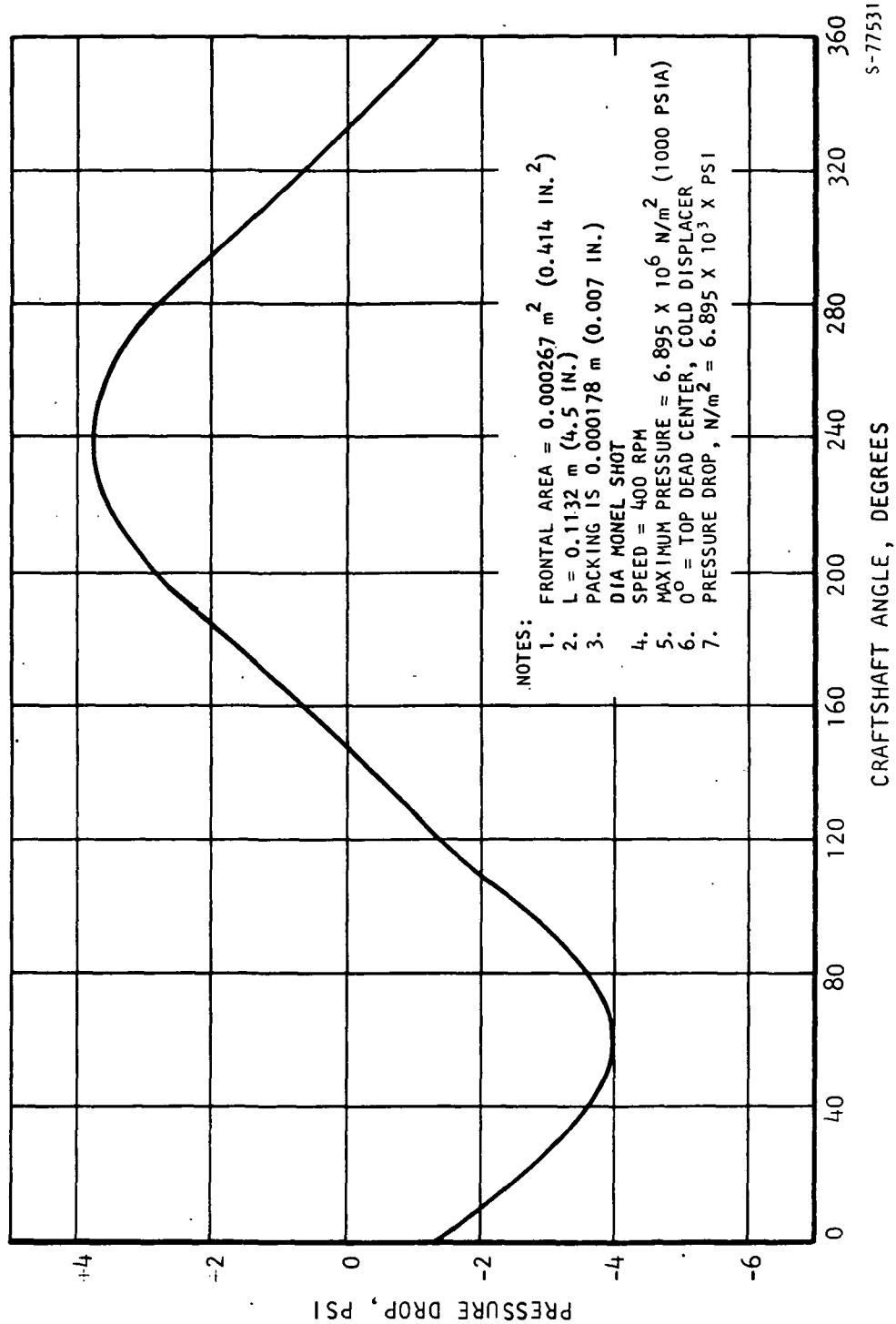


Figure 3-4. Pressure Drop of First Iteration Cold Regenerator

The metal and gas temperatures at the cold end of this second iteration regenerator are presented in Figure 3-5. Figure 3-6 presents the gas pressure drop characteristic. The maximum temperature deviation is reduced to 0.855°K (1.54°R), and maximum pressure drop is $1.048 \times 10^4 \text{ N/m}^2$ (1.52 psi). These values represent considerable improvement over the first design iteration. However, when all losses were considered, it was decided that the remaining net refrigeration with this configuration was too marginal for this early stage in the program. The regenerator was accordingly lengthened, to reduce conduction losses, thus leading to the selected preliminary design.

4. Selected Design

The cold regenerator selected for the preliminary design fractional watt VM is of an annular configuration, packed with two sizes of spherical monel shot. Monel was chosen as the material because of its superior heat capacity in the temperature range of interest (see Appendix A). The total length is 0.127m (5 in.), and the annular height is 0.00762m (0.3 in.), 30 diameters of the largest shot, in order to prevent gas channeling along the walls. The first 0.0762m (3 in.) from the sump end are packed with 0.000354m (0.010 in.) diameter shot; the remaining .0508m (2 in.) are packed with 0.000178m (0.007 in.) diameter shot. The frontal area is 0.00044m^2 (0.682 in.^2).

The first section of the regenerator, 0.000254m (0.010 in.) dia shot, has a porosity of 39 percent, an area to volume ratio of $14,400\text{m}^2/\text{m}^3$ ($366 \text{ in.}^2/\text{in.}^3$), and a hydraulic diameter of 0.0001086 (0.004274 in.). For the second section 0.000178m (0.007 in.) dia. shot, the porosity is 39 percent, the area to volume ratio is $20,600\text{m}^2/\text{m}^3$ ($523 \text{ in.}^2/\text{in.}^3$), and the hydraulic diameter is 0.0000759m (0.002988 in.).

The larger diameter shot is used in the warm end of the cold regenerator to reduce the pressure drop, but still maintain a large thermal capacity. The smaller diameter shot at the cold end yields the required high heat transfer coefficient.

Table 3-7 presents the detailed output from the regenerator analysis computer program for this preliminary design cold regenerator. The nodewise pressures and temperatures of the gas are listed as a function of the angular position of the crankshaft, or time. Angular displacement is referenced to the top dead center position of the cold displacer. The output parameters are matrix temperature, gas temperature, gas density, gas pressure, and gas flow rate. Positive gas flow rate denotes mass flow from the sump end of the regenerator toward the cold end. Node 0 represents the sump end of the regenerator, and Node 13 the outlet face at the cold end. Nodes 1 through 7 are the portion of the core packed with 0.000254m (0.010 in.) dia shot, and Nodes 8 through 12 represent the 0.000178m (0.007 in.) dia shot.

Figures 3-7 through 3-9 present plots of key parameters from the computer program printout. The matrix and gas temperatures at the cold end of the regenerator are plotted in Figure 3-7. The small difference between the two temperatures is indicative of excellent heat transfer, and the 0.889°K (1.6°R) temperature swing of the gas is indicative of adequate heat capacity.



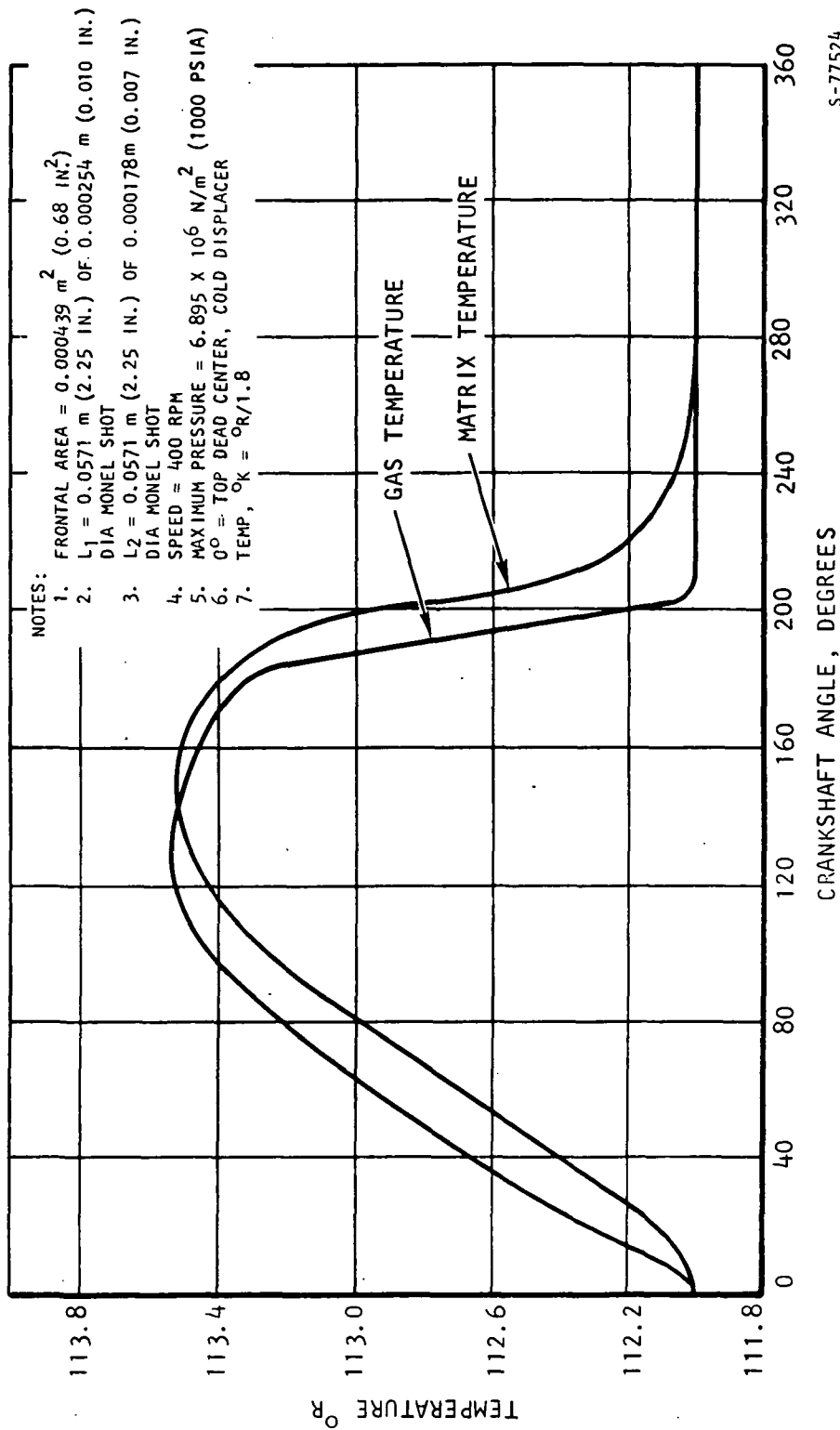


Figure 3-5. Temperature Variation at Cold End of Second Iteration Cold Regenerator

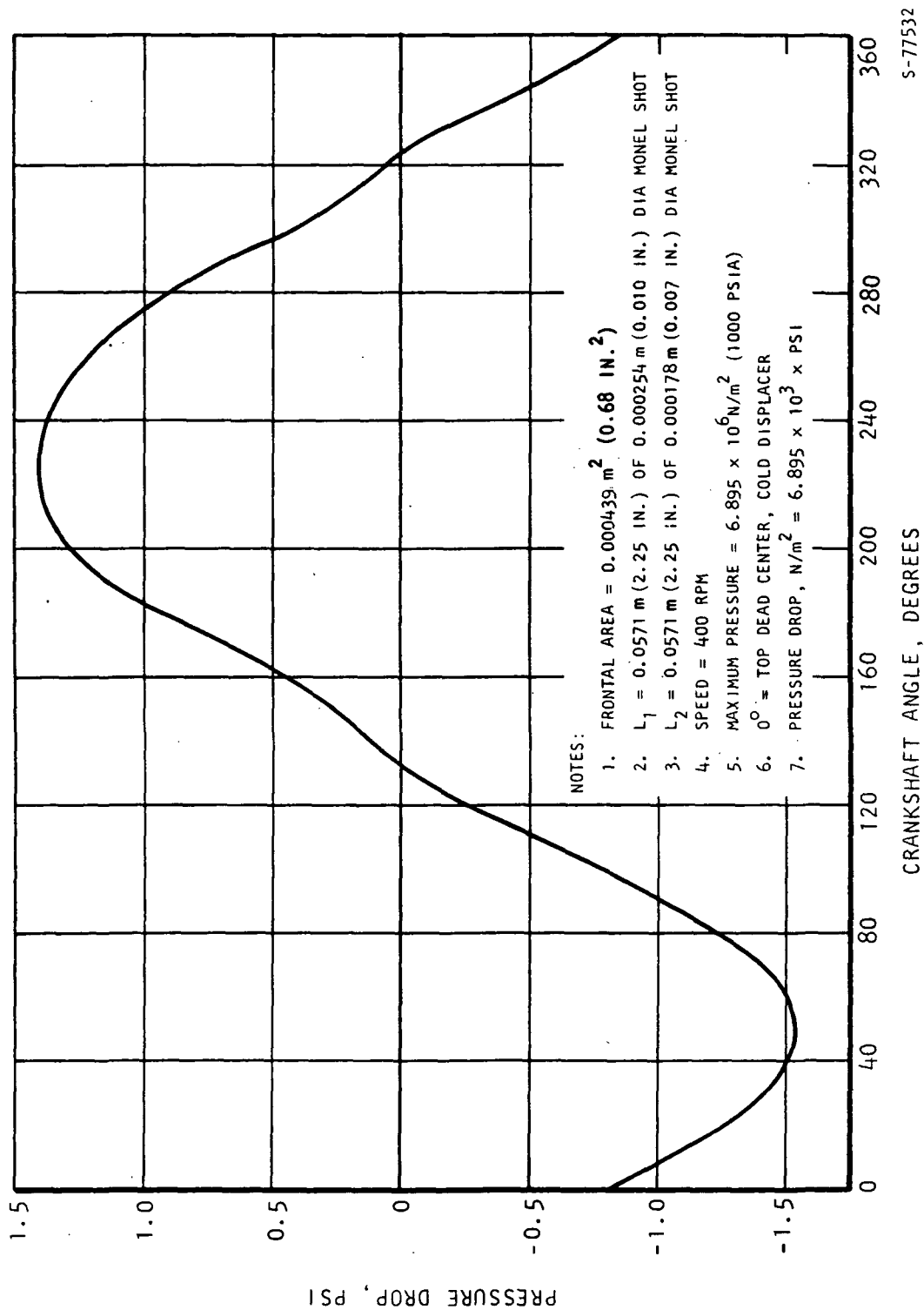


Figure 3-6. Pressure Drop of Second Iteration Cold Regenerator

TABLE 3-7

PRELIMINARY DESIGN
COLD REGENERATOR
PERFORMANCE CHARACTERISTICS

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
$\theta = 0^\circ$					
REGULAR PRINTOUTS					
TIME(SEC.) = 4.49989-01			DATE = 27 SEP 72 TIME = 17119133		
			4722 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19488+02	6.20000+02	5.53459-01	9.50534+02	2.17912-03
1	5.96764+02	5.97085+02	5.73584-01	9.50408+02	2.13282-03
2	5.56505+02	5.57110+02	6.14103-01	9.50185+02	2.03389-03
3	5.14275+02	5.14913+02	6.62202-01	9.49997+02	1.92747-03
4	4.71956+02	4.72609+02	7.19777-01	9.49841+02	1.81214-03
5	4.29841+02	4.30309+02	7.88209-01	9.49716+02	1.68630-03
6	3.87336+02	3.88019+02	8.68515-01	9.49617+02	1.54818-03
7	3.45036+02	3.45736+02	9.49237-01	9.49542+02	1.39484-03
8	3.02796+02	3.03211+02	1.09818+00	9.49481+02	1.22221-03
9	2.60416+02	2.60828+02	1.26760+00	9.49419+02	1.03780-03
10	2.18133+02	2.18563+02	1.49931+00	9.49380+02	8.22114-04
11	1.76017+02	1.76472+02	1.83578+00	9.49360+02	5.62211-04
12	1.35948+02	1.36472+02	2.33832+00	9.49352+02	2.38966-04
13	1.12002+02	1.12119+02	2.80965+00	9.49351+02	4.88867-05

$\theta = 30^\circ$					
TIME(SEC.) = 4.62479-01			DATE = 27 SEP 72 TIME = 17119151		
			4868 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19839+02	6.20000+02	5.69082-01	9.78303+02	2.73966-03
1	5.96980+02	5.97310+02	5.89514-01	9.78136+02	2.70238-03
2	5.56900+02	5.57516+02	6.30860-01	9.77834+02	2.62275-03
3	5.14683+02	5.15339+02	6.80132-01	9.77573+02	2.53711-03
4	4.72363+02	4.73039+02	7.39064-01	9.77350+02	2.44433-03
5	4.30046+02	4.30740+02	8.09166-01	9.77163+02	2.34311-03
6	3.87737+02	3.88450+02	8.91340-01	9.77010+02	2.23208-03
7	3.45437+02	3.46170+02	9.94387-01	9.76887+02	2.10883-03
8	3.03227+02	3.03664+02	1.12625+00	9.76780+02	1.97016-03
9	2.60841+02	2.61280+02	1.29933+00	9.76658+02	1.82213-03
10	2.18580+02	2.19038+02	1.53568+00	9.76571+02	1.64918-03
11	1.76502+02	1.76979+02	1.87815+00	9.76514+02	1.44108-03
12	1.36511+02	1.36988+02	2.38827+00	9.76479+02	1.18285-03
13	1.12340+02	1.12594+02	2.86844+00	9.76469+02	1.03120-03

- NOTES: 1. TEMP, $^{\circ}\text{K} = ^{\circ}\text{R}/1.8$
 2. DENSITY, $\text{kg}/\text{m}^3 = 16.02 \times \text{LB}/\text{FT}^3$
 3. PRESSURE, $\text{N}/\text{m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg}/\text{sec} = 0.4536 \times \text{LB}/\text{SEC}$



TABLE 3-7 (Continued)

NODE MATRIX GAS GAS GAS GAS
NO. TEMPERATURE TEMPERATURE DENSITY PRESSURE FLOW
RATE

 $\theta = 60^\circ$

REGULAR PRINTOUTS

DATE = 27 SEP 72 TIME = 17120111

TIME(SEC.) = 4.74931-01

5013 = COUNT (NO. OF CALCULATIONS)

N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19917+02	6.20000+02	5.79333-01	9.96553+02	2.52277-03
1	5.97183+02	5.97460+02	5.99985-01	9.96406+02	2.50146-03
2	5.57268+02	5.57781+02	6.41923-01	9.96137+02	2.45595-03
3	5.15069+02	5.15622+02	6.92012-01	9.95900+02	2.40700-03
4	4.72753+02	4.73325+02	7.51881-01	9.95694+02	2.35400-03
5	4.30438+02	4.31029+02	8.23126-01	9.95518+02	2.29619-03
6	3.88131+02	3.88742+02	9.06569-01	9.95370+02	2.23278-03
7	3.45837+02	3.46466+02	1.01118+00	9.95247+02	2.16244-03
8	3.03664+02	3.04042+02	1.14477+00	9.95136+02	2.08333-03
9	2.61285+02	2.61673+02	1.32017+00	9.95004+02	1.99893-03
10	2.19059+02	2.19467+02	1.55929+00	9.94905+02	1.90038-03
11	1.77041+02	1.77471+02	1.90494+00	9.94833+02	1.78199-03
12	1.37156+02	1.37588+02	2.41715+00	9.94785+02	1.63545-03
13	1.12798+02	1.13047+02	2.90462+00	9.94769+02	1.54938-03

 $\theta = 90^\circ$

DATE = 27 SEP 72 TIME = 17120129

TIME(SEC.) = 4.87426-01

5147 = COUNT (NO. OF CALCULATIONS)

N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19956+02	6.20000+02	5.80381-01	9.98419+02	1.47802-03
1	5.97309+02	5.97433+02	6.01140-01	9.98348+02	1.48822-03
2	5.57494+02	5.57712+02	6.43289-01	9.98213+02	1.50999-03
3	5.15312+02	5.15560+02	6.93549-01	9.98088+02	1.53340-03
4	4.73004+02	4.73268+02	7.53617-01	9.97974+02	1.55876-03
5	4.30695+02	4.30976+02	8.25074-01	9.97871+02	1.58641-03
6	3.88395+02	3.88697+02	9.08751-01	9.97779+02	1.61675-03
7	3.46112+02	3.46436+02	1.01361+00	9.97698+02	1.65041-03
8	3.03971+02	3.04172+02	1.14699+00	9.97618+02	1.68824-03
9	2.61614+02	2.61835+02	1.32250+00	9.97515+02	1.72857-03
10	2.19426+02	2.19671+02	1.56157+00	9.97430+02	1.77565-03
11	1.77474+02	1.77747+02	1.90649+00	9.97361+02	1.83218-03
12	1.37499+02	1.37983+02	2.41563+00	9.97308+02	1.90209-03
13	1.13239+02	1.13432+02	2.90239+00	9.97287+02	1.94313-03

- NOTES: 1. TEMP, $^\circ\text{K} = ^\circ\text{R}/1.8$
 2. DENSITY, $\text{kg/m}^3 = 16.02 \times \text{LB/FT}^3$
 3. PRESSURE, $\text{N/m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg/sec} = 0.4536 \times \text{LB/SEC}$



TABLE 3-7 (Continued)

$\theta = 120^\circ$

REGULAR PRINTOUTS

DATE = 27 SEP 72 TIME = 17:25:42
5250 = COUNT (NO. OF CALCULATIONS)

TIME(SEC.) = 4.99999-01

N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19972+02	6.20000+02	5.72329-01	9.84081+02	8.44207-08
1	5.97322+02	5.97155+02	5.93124-01	9.84081+02	2.94475-05
2	5.57510+02	5.57231+02	6.35056-01	9.84079+02	9.21793-05
3	5.15340+02	5.15103+02	6.84771-01	9.84075+02	1.59634-04
4	4.73039+02	4.72835+02	7.44244-01	9.84069+02	2.32711-04
5	4.30738+02	4.30563+02	8.14926-01	9.84060+02	3.12428-04
6	3.88448+02	3.88303+02	8.97767-01	9.84048+02	3.99871-04
7	3.46177+02	3.46061+02	1.00156+00	9.84034+02	4.96893-04
8	3.04053+02	3.04003+02	1.13291+00	9.84016+02	6.05914-04
9	2.61723+02	2.61704+02	1.30644+00	9.83988+02	7.22239-04
10	2.19566+02	2.19572+02	1.54284+00	9.83959+02	8.58020-04
11	1.77664+02	1.77701+02	1.88375+00	9.83931+02	1.02111-03
12	1.37967+02	1.38034+02	2.38615+00	9.83905+02	1.22279-03
13	1.13518+02	1.13605+02	2.86499+00	9.83893+02	1.34116-03

$\theta = 150^\circ$

DATE = 27 SEP 72 TIME = 17:25:55
5355 = COUNT (NO. OF CALCULATIONS)

TIME(SEC.) = 5.12496-01

N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19870+02	6.19606+02	5.57844-01	9.57720+02	-1.38458-03
1	5.97133+02	5.96639+02	5.78315-01	9.57787+02	-1.33932-03
2	5.57325+02	5.56811+02	6.19235-01	9.57904+02	-1.24262-03
3	5.15162+02	5.14641+02	6.67894-01	9.57998+02	-1.13860-03
4	4.72869+02	4.72340+02	7.26160-01	9.58072+02	-1.02588-03
5	4.30578+02	4.30042+02	7.95339-01	9.58128+02	-9.02898-04
6	3.88304+02	3.87759+02	8.76519-01	9.58168+02	-7.67927-04
7	3.46048+02	3.45494+02	9.78252-01	9.58196+02	-6.18082-04
8	3.03928+02	3.03616+02	1.10631+00	9.58213+02	-4.49737-04
9	2.61627+02	2.61299+02	1.27641+00	9.58227+02	-2.69993-04
10	2.19477+02	2.19395+02	1.50683+00	9.58232+02	-6.01729-05
11	1.77566+02	1.77265+02	1.84326+00	9.58231+02	1.92377-04
12	1.37896+02	1.37681+02	2.33695+00	9.58225+02	5.05190-04
13	1.13563+02	1.13512+02	2.80223+00	9.58221+02	6.88703-04

- NOTES: 1. TEMP, $^{\circ}\text{K} = ^{\circ}\text{R}/1.8$
 2. DENSITY, $\text{kg}/\text{m}^3 = 16.02 \times \text{LB}/\text{FT}^3$
 3. PRESSURE, $\text{N}/\text{m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg}/\text{sec} = 0.4536 \times \text{LB}/\text{SEC}$



TABLE 3-7 (Continued)

$\theta = 180^\circ$

REGULAR PRINTOUTS

DATE = 27 SEP 72 TIME = 17121133
5491 = COUNT (NO. OF CALCULATIONS)

TIME(SEC.) = 5.24932-01

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LB/M/CF	PG(N) PSIA	WG(N) LB/M/SEC
-0	6.19484+02	6.19368+02	5.41404-01	9.28209+02	-2.29608-03
1	5.96803+02	5.96218+02	5.61522-01	9.28347+02	-2.25073-03
2	5.56990+02	5.56379+02	6.01417-01	9.28592+02	-2.15383-03
3	5.14832+02	5.14205+02	6.48823-01	9.28800+02	-2.04958-03
4	4.72544+02	4.71906+02	7.05648-01	9.28973+02	-1.93657-03
5	4.30261+02	4.29613+02	7.73051-01	9.29114+02	-1.81324-03
6	3.87995+02	3.87336+02	8.52224-01	9.29226+02	-1.67784-03
7	3.45748+02	3.45080+02	9.51436-01	9.29312+02	-1.52746-03
8	3.03627+02	3.03249+02	1.07618+00	9.29382+02	-1.35445-03
9	2.61335+02	2.60951+02	1.24218+00	9.29456+02	-1.17788-03
10	2.19206+02	2.18817+02	1.46874+00	9.29504+02	-9.66729-04
11	1.77351+02	1.76967+02	1.79615+00	9.29531+02	-7.12516-04
12	1.37718+02	1.37507+02	2.27769+00	9.29543+02	-3.97321-04
13	1.13131+02	1.12000+02	2.76124+00	9.29545+02	-2.10433-04

$\theta = 210^\circ$

DATE = 27 SEP 72 TIME = 17121133
5645 = COUNT (NO. OF CALCULATIONS)

TIME(SEC.) = 5.37500-01

N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LB/M/CF	PG(N) PSIA	WG(N) LB/M/SEC
-0	6.19479+02	6.19174+02	5.27374-01	9.03100+02	-2.54000-03
1	5.96429+02	5.95880+02	5.47148-01	9.03263+02	-2.50796-03
2	5.56604+02	5.56022+02	5.86124-01	9.03557+02	-2.43950-03
3	5.14449+02	5.13852+02	6.32418-01	9.03812+02	-2.36582-03
4	4.72162+02	4.71552+02	6.87967-01	9.04030+02	-2.28595-03
5	4.29883+02	4.29258+02	7.53815-01	9.04214+02	-2.19875-03
6	3.87621+02	3.86982+02	8.31234-01	9.04365+02	-2.10300-03
7	3.45376+02	3.44723+02	9.28268-01	9.04487+02	-1.99663-03
8	3.03242+02	3.02864+02	1.05036+00	9.04594+02	-1.87700-03
9	2.60945+02	2.60558+02	1.21296+00	9.04716+02	-1.74911-03
10	2.18803+02	2.18405+02	1.43517+00	9.04805+02	-1.59941-03
11	1.76936+02	1.76543+02	1.75673+00	9.04864+02	-1.41897-03
12	1.37453+02	1.37218+02	2.22858+00	9.04901+02	-1.19506-03
13	1.12254+02	1.12000+02	2.69746+00	9.04912+02	-1.06237-03

- NOTES: 1. TEMP, $^{\circ}\text{K} = ^{\circ}\text{R}/1.8$
 2. DENSITY, $\text{kg/m}^3 = 16.02 \times \text{LB/FT}^3$
 3. PRESSURE, $\text{N/m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg/sec} = 0.4536 \times \text{LB/SEC}$



TABLE 3-7 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 240^\circ$					
TIME(SEC.) = 5.49977-01			DATE = 27 SEP 72 TIME = 17:21:52 5880 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19298+02	6.19048+02	5.18945-01	8.88030+02	-2.20096-03
1	5.96108+02	5.95666+02	5.38487-01	8.88167+02	-2.18364-03
2	5.56269+02	5.55796+02	5.76865-01	8.88416+02	-2.14662-03
3	5.14112+02	5.13624+02	6.22442-01	8.88636+02	-2.10679-03
4	4.71824+02	4.71322+02	6.77177-01	8.88826+02	-2.06359-03
5	4.29544+02	4.29027+02	7.42040-01	8.88990+02	-2.01643-03
6	3.87281+02	3.86748+02	8.18357-01	8.89128+02	-1.96463-03
7	3.45033+02	3.44483+02	9.14037-01	8.89242+02	-1.90706-03
8	3.02875+02	3.02551+02	1.03465+00	8.89346+02	-1.84231-03
9	2.60566+02	2.60230+02	1.19522+00	8.89469+02	-1.77305-03
10	2.18398+02	2.18047+02	1.41495+00	8.89563+02	-1.69191-03
11	1.76504+02	1.76154+02	1.73338+00	8.89630+02	-1.59401-03
12	1.37139+02	1.36916+02	2.19997+00	8.89675+02	-1.47243-03
13	1.12043+02	1.12000+02	2.65770+00	8.89690+02	-1.40045-03

$\theta = 270^\circ$					
TIME(SEC.) = 5.62500-01			DATE = 27 SEP 72 TIME = 17:22:12 5946 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19184+02	6.19062+02	5.18137-01	8.86610+02	-1.35500-03
1	5.95912+02	5.95706+02	5.37579-01	8.86681+02	-1.36314-03
2	5.56060+02	5.55829+02	5.75827-01	8.86815+02	-1.38052-03
3	5.13898+02	5.13654+02	6.21263-01	8.86938+02	-1.39923-03
4	4.71604+02	4.71345+02	6.75844-01	8.87050+02	-1.41952-03
5	4.29320+02	4.29044+02	7.40542-01	8.87151+02	-1.44167-03
6	3.87051+02	3.86757+02	8.16682-01	8.87241+02	-1.46601-03
7	3.44793+02	3.44478+02	9.12175-01	8.87321+02	-1.49306-03
8	3.02607+02	3.02410+02	1.03295+00	8.87398+02	-1.52350-03
9	2.60279+02	2.60065+02	1.19345+00	8.87496+02	-1.55606-03
10	2.18077+02	2.17841+02	1.41328+00	8.87578+02	-1.59421-03
11	1.76148+02	1.75902+02	1.73221+00	8.87643+02	-1.64028-03
12	1.36843+02	1.36664+02	2.19966+00	8.87693+02	-1.69754-03
13	1.12004+02	1.12000+02	2.65253+00	8.87712+02	-1.73138-03

- NOTES: 1. TEMP, $^\circ K = ^\circ R / 1.8$
 2. DENSITY, $kg/m^3 = 16.02 \times LB/FT^3$
 3. PRESSURE, $N/m^2 = 6.895 \times 10^3 \times PSI$
 4. FLOW, $kg/sec = 0.4536 \times LB/SEC$



TABLE 3-7 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
$\theta = 300^\circ$					
REGULAR PRINTOUTS					
DATE = 27 SEP 72 TIME = 17:22:127					
TIME(SEC.) = 5.74999-01					
6083 = COUNT (NO. OF CALCULATIONS)					
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19162+02	6.19229+02	5.24545-01	8.98179+02	-2.00107-04
1	5.95581+02	5.96017+02	5.44046-01	8.98184+02	-2.24369-04
2	5.56019+02	5.56132+02	5.82654-01	8.98197+02	-2.76223-04
3	5.13851+02	5.13946+02	6.28535-01	8.98212+02	-3.32020-04
4	4.71550+02	4.71626+02	6.83620-01	8.98229+02	-3.92522-04
5	4.29257+02	4.29313+02	7.48965-01	8.98248+02	-4.58576-04
6	3.86979+02	3.87012+02	8.25828-01	8.98268+02	-5.31126-04
7	3.44710+02	3.44716+02	9.22252-01	8.98290+02	-6.11750-04
8	3.02500+02	3.02480+02	1.04471+00	8.98314+02	-7.02505-04
9	2.60151+02	2.60111+02	1.20690+00	8.98351+02	-7.99583-04
10	2.17917+02	2.17851+02	1.42914+00	8.98385+02	-9.13294-04
11	1.75952+02	1.75863+02	1.75172+00	8.98418+02	-1.05057-03
12	1.36644+02	1.36549+02	2.22531+00	8.98446+02	-1.22117-03
13	1.12001+02	1.12000+02	2.68059+00	8.98459+02	-1.32185-03

$\theta = 330^\circ$ DATE = 27 SEP 72 TIME = 17:22:139
TIME(SEC.) = 5.87485-01 6161 = COUNT (NO. OF CALCULATIONS)

N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19422+02	6.20000+02	5.37015-01	9.21362+02	1.06847-03
1	5.95971+02	5.96219+02	5.57406-01	9.21313+02	1.02641-03
2	5.56169+02	5.56639+02	5.96539-01	9.21250+02	9.36590-04
3	5.13996+02	5.14489+02	6.43354-01	9.21164+02	8.39951-04
4	4.71688+02	4.72193+02	6.99497-01	9.21114+02	7.35199-04
5	4.29384+02	4.29906+02	7.66163-01	9.21077+02	6.20871-04
6	3.87096+02	3.87635+02	8.44450-01	9.21053+02	4.95363-04
7	3.44814+02	3.45384+02	9.42602-01	9.21038+02	3.55955-04
8	3.02587+02	3.02954+02	1.06804+00	9.21030+02	1.98985-04
9	2.60212+02	2.60725+02	1.23265+00	9.21027+02	3.12932-05
10	2.17956+02	2.18183+02	1.46033+00	9.21028+02	-1.65193-04
11	1.75963+02	1.76108+02	1.78948+00	9.21033+02	-4.02253-04
12	1.36581+02	1.36594+02	2.27432+00	9.21044+02	-6.96765-04
13	1.12000+02	1.12000+02	2.73928+00	9.21050+02	-8.70446-04

- NOTES: 1. TEMP, $^\circ K = ^\circ R / 1.8$
 2. DENSITY, $kg/m^3 = 16.02 \times LB/FT^3$
 3. PRESSURE, $N/m^2 = 6.895 \times 10^3 \times PSI$
 4. FLOW, $kg/sec = 0.4536 \times LB/SEC$



TABLE 3-7 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
$\theta = 360^\circ$					
REGULAR PRINTOUTS					
DATE = 27 SEP 72 TIME = 17:22:57			6296 = COUNT (NO. OF CALCULATIONS)		
TIME(SEC.) = 5.99988-01					
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.19682+02	6.20000+02	5.53458-01	9.50532+02	2.17906-03
1	5.96142+02	5.96469+02	5.74173-01	9.50406+02	2.13272-03
2	5.56477+02	5.57076+02	6.14139-01	9.50184+02	2.03380-03
3	5.14310+02	5.14947+02	6.62160-01	9.49995+02	1.92739-03
4	4.71999+02	4.72651+02	7.19712-01	9.49840+02	1.81208-03
5	4.29690+02	4.30358+02	7.88124-01	9.49714+02	1.68625-03
6	3.87394+02	3.88077+02	8.68387-01	9.49615+02	1.54817-03
7	3.45106+02	3.45806+02	9.69044-01	9.49540+02	1.39486-03
8	3.02894+02	3.03309+02	1.09785+00	9.49480+02	1.22231-03
9	2.60504+02	2.60915+02	1.26720+00	9.49417+02	1.03797-03
10	2.18240+02	2.18670+02	1.49863+00	9.49379+02	8.22394-04
11	1.76232+02	1.76687+02	1.83366+00	9.49358+02	5.62784-04
12	1.36724+02	1.37244+02	2.32516+00	9.49350+02	2.41133-04
13	1.12002+02	1.12136+02	2.80929+00	9.49349+02	5.10956-05

- NOTES: 1. TEMP, $^{\circ}\text{K} = ^{\circ}\text{R}/1.8$
 2. DENSITY, $\text{kg}/\text{m}^3 = 16.02 \times \text{LB}/\text{FT}^3$
 3. PRESSURE, $\text{N}/\text{m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg}/\text{sec} = 0.4536 \times \text{LB}/\text{SEC}$



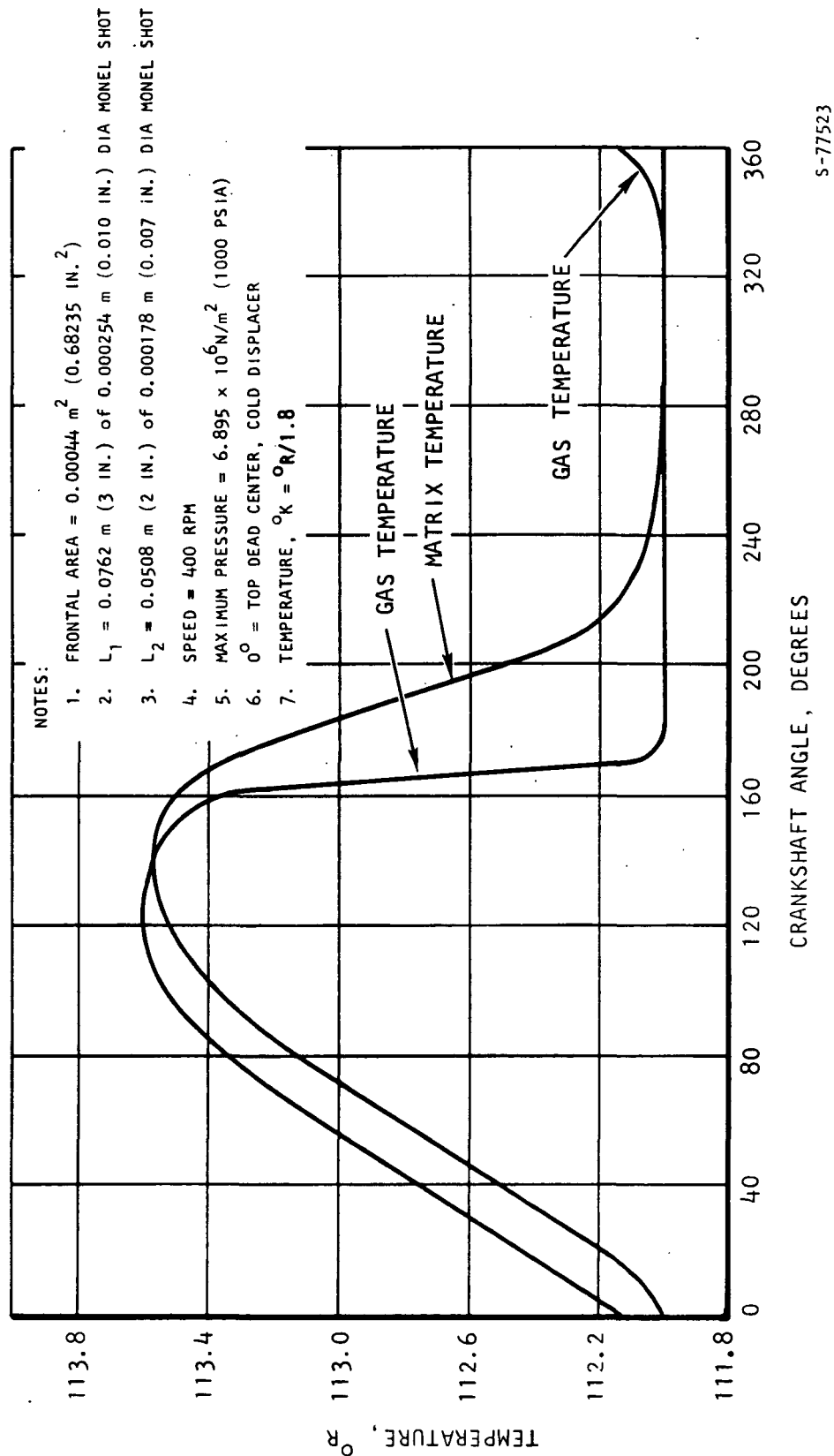


Figure 3-7. Temperature Variation at Cold End of Preliminary Design Cold Regenerator

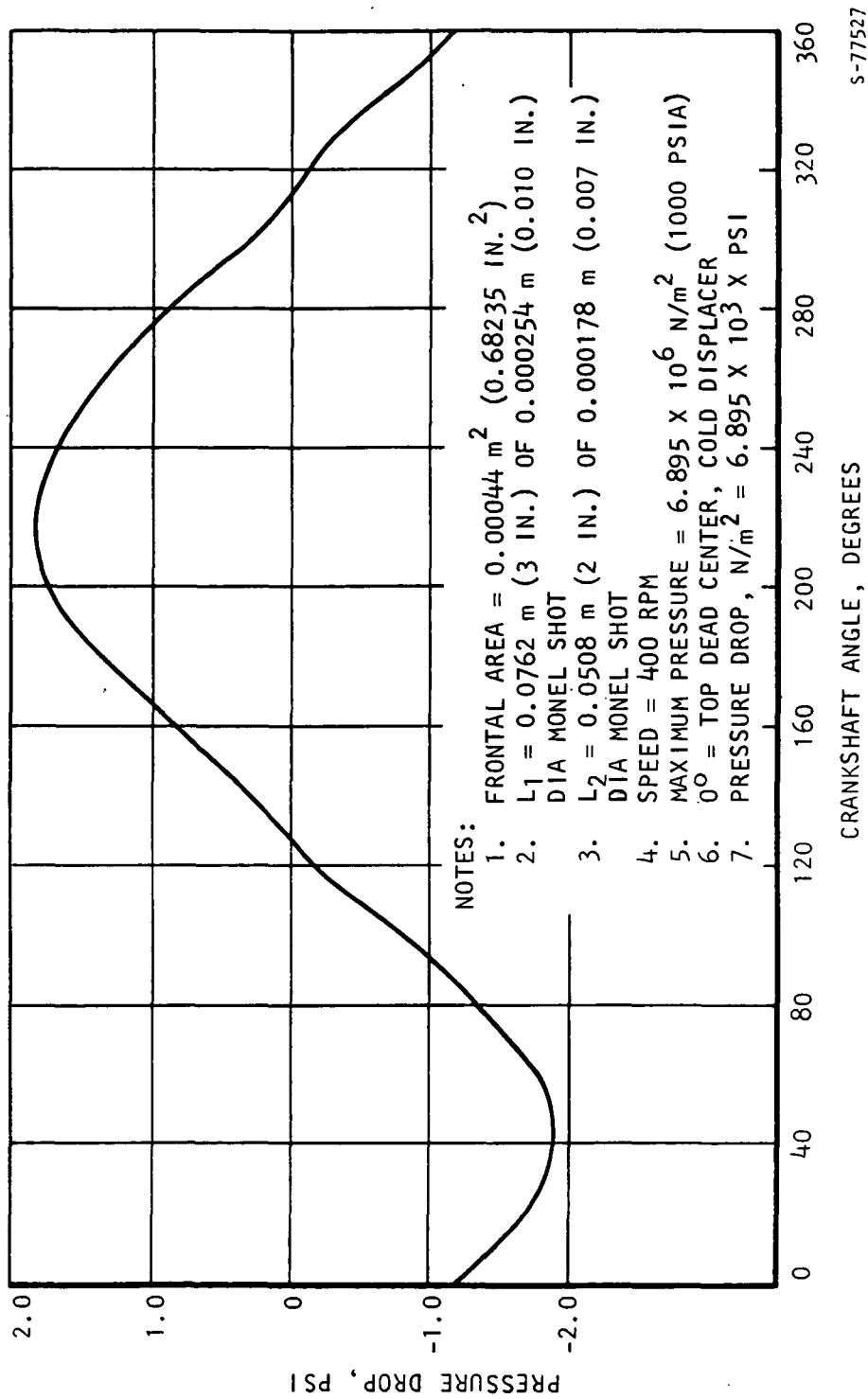


Figure 3-8. Pressure Drop of Preliminary Design Cold Regenerator

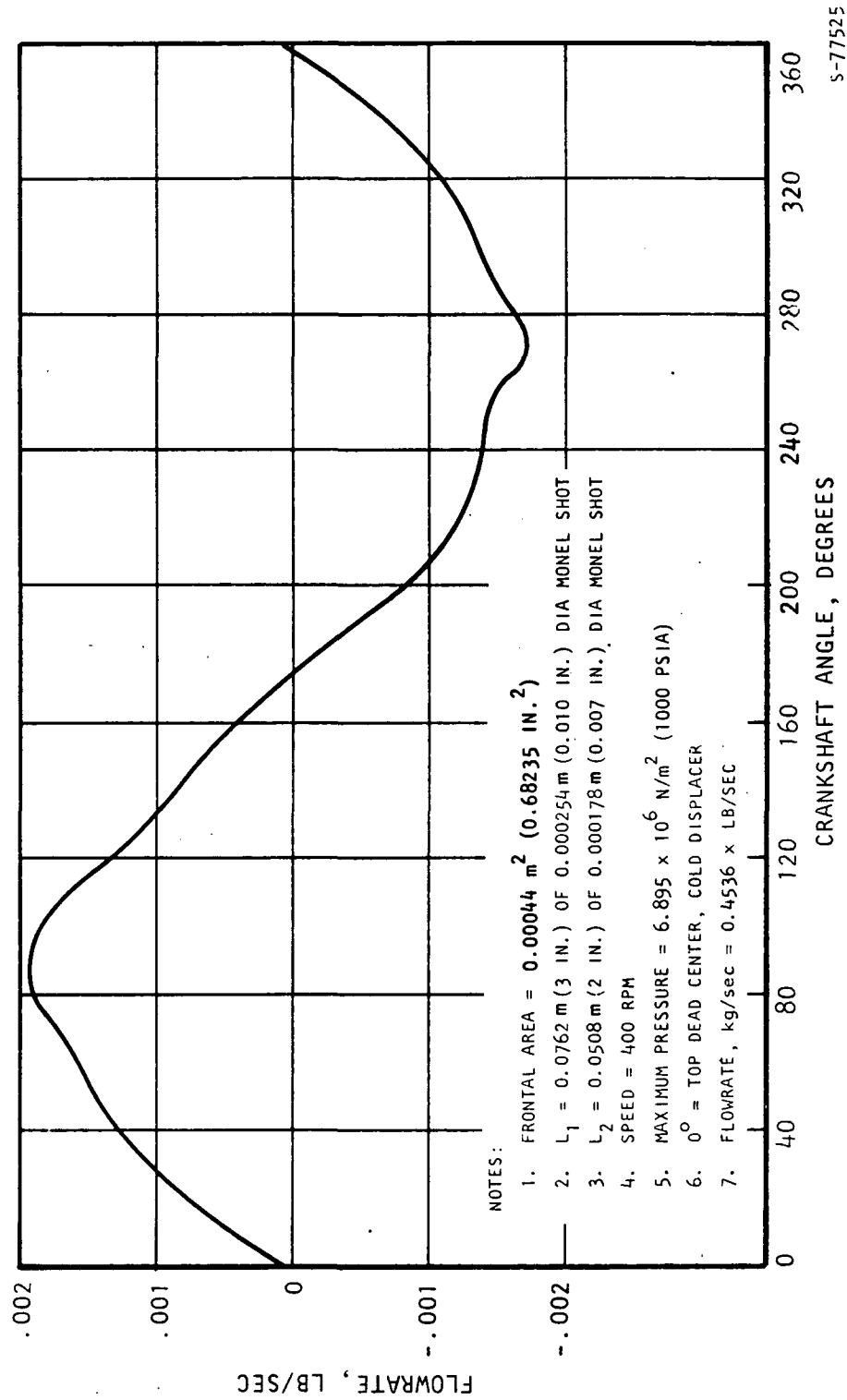


Figure 3-9. Cold End Flow Rate of Preliminary Design Cold Regenerator



Figure 3-8 indicates a maximum regenerator pressure drop of $1.26 \times 10^4 \text{ N/m}^2$ (1.83 psi). This pressure drop, coupled with the low ΔP of the remainder of the components in the cold gas flow path, will limit leakage rates past the displacer to very low values. The regenerator represents the major portion of the cold gas pressure drop. However, attempts to decrease pressure drops would lead to losses in thermal performance which would impose a greater overall penalty on the VM. Figure 3-9 presents the gas flow rate at the cold end of the regenerator. This plot, along with the gas temperature, is used to determine the regenerator losses.

An ideal regenerator will always supply gas to the cold end of the VM at a constant temperature which is equal to that of the cold expansion volume temperature. In the actual case, this temperature varies as a result of the finite matrix heat capacity and heat transfer coefficient between the gas and matrix. This is evidenced by the temperature plots of Figure 3-7. The losses associated with this departure from ideal conditions are estimated by integrating the excess fluid energy supplied to the cold end over a complete cycle, or crankshaft revolution. The loss per cycle is expressed as:

$$\dot{Q}_{\text{loss}} = \int_0^{2\pi} \dot{w} \text{ cp } (T - T_{\text{ref}}) d\theta \quad (3-1)$$

Where

$\dot{Q}_{\text{loss}}$ = regenerator loss per revolution

$\dot{w}$ = fluid flow rate

cp = specific heat (function of temperature and pressure)

T = fluid temperature

T_{ref} = cold expansion volume temperature

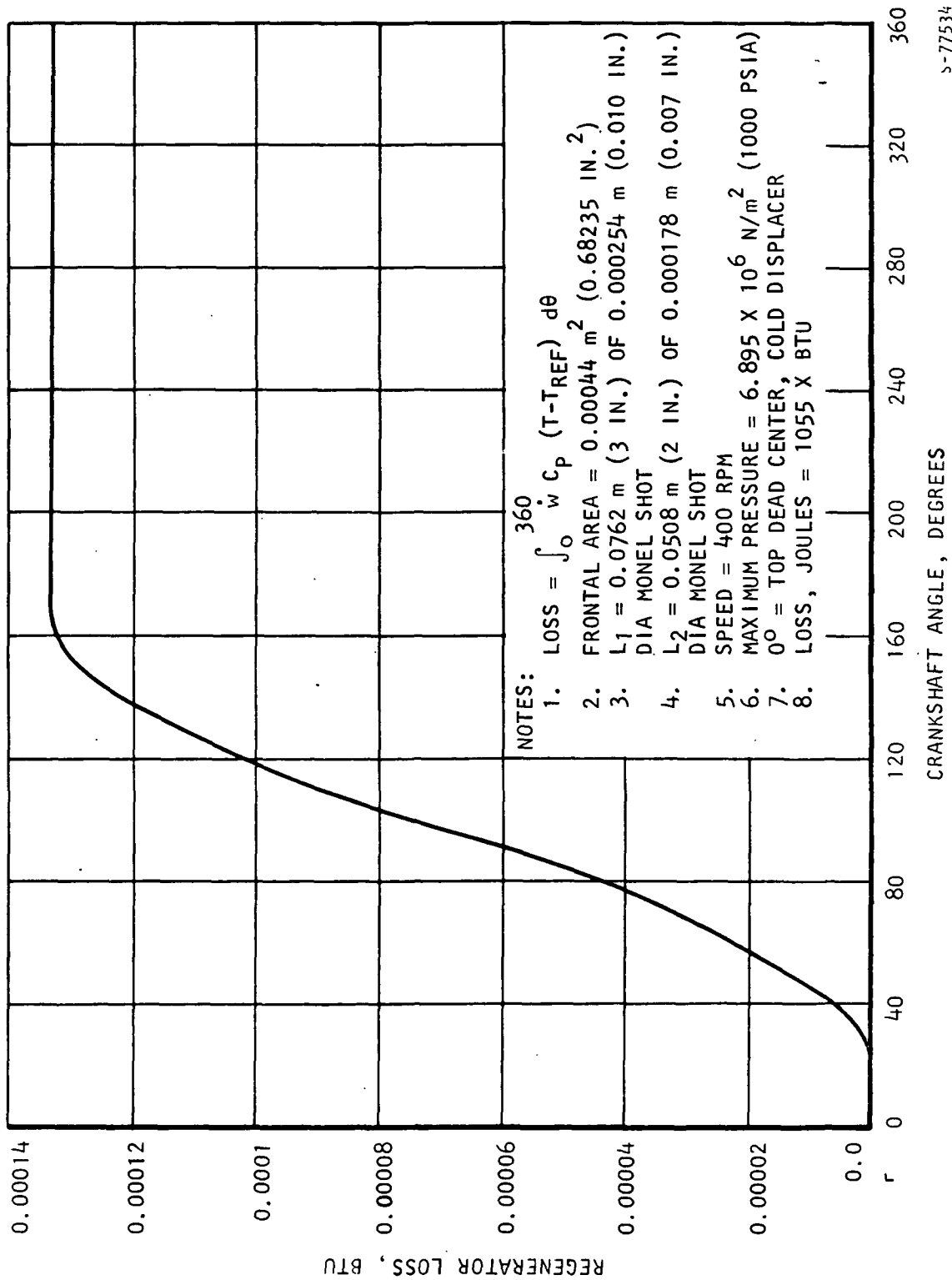
Figures 3-7 and 3-9, along with the specific heat characteristics of helium, allow a stepwise evaluation of Equation 3-1. The integrated loss as a function of crank angle is presented in Figure 3-10. The total loss per cycle is 0.1402J (0.0001329 Btu). At a rotational speed of 400 rpm, this loss translates to a total of 0.929 watts.

Hot Regenerator Design

1. Introduction

The hot regenerator serves as a thermal isolator between the sump and the high temperature portions of the VM refrigerator. As the gas flows from the sump towards the hot end, it is heated to a temperature approaching the maximum in the cycle. At the time of flow reversal, an expansion process begins which allows heat to be added to the gas at essentially constant temperature. The expanding gas is cooled as it flows through the regenerator, thus storing energy in the regenerator matrix and reestablishing the temperature profile for the next half. Desirable design features are similar to those of cold regenerator, with major emphasis placed on different items.





S-77534

Figure 3-10. Preliminary Design Cold Regenerator Thermal Loss per Cycle

The hot displacer is considerably larger than the cold, and thus the pressure differential across the displacer strongly influences motor drive power. Thus regenerator pressure loss becomes an item of prime importance. Thermal efficiency assumes a less important role in the hot regenerator than in the cold. The thermal losses at the hot end are compensated for by additional heat input on a one to one basis, while at the cold end the thermal power input to offset the losses is magnified by the coefficient of performance of the refrigerator. Void volume at the higher temperature does not affect performance nearly as strongly as at colder temperatures, since the mass of gas stored in the high temperature region is smaller, due to the high temperature.

2. Method of Analysis

The hot regenerator is analyzed in the same manner as the cold. A description of the analytical methods used in the computer program, along with the physical characteristics of candidate packings, is presented in Appendix A of this report.

3. Initial Iterations

The experience gained in the design iterations on the cold regenerator and from the analysis performed on the GSFC 5 watt VM allowed rapid convergence upon the configuration selected for the preliminary design hot regenerator. It was originally planned to utilize a hot regenerator packing composed at least partially of spherical shot. However, ratioing of the values obtained for the cold regenerator indicated that the pressure drop would be too high, as would the conduction losses from the hot to the sump region. Order of magnitude pressure drops were obtained by assuming that pressure drop varied directly with the square of mass velocity (flow per unit free flow area) and inversely with fluid density. The estimates of regenerator conduction losses were obtained by selecting a reasonable regenerator void volume, applying the ground rule of a 30 shot diameter annular height, and thus determining a regenerator length. With this type of estimate, it was determined that the sum of the thermal conduction losses of the regenerator walls and packing, and the idealized heat input was approximately equal to the allowable heat input. Since other factors also contribute to the heat input, this was not an acceptable solution. In addition, the estimated maximum pressure drops for these regenerators ranged from 1.103×10^4 (1.6) to $4.55 \times 10^4 \text{ N/m}^2$ (6.6 psi), which would have resulted in excessive drive motor power requirements.

The results described above led to the decision to utilize screen as the hot regenerator packing material. The experience gained from the design of the screen packed hot regenerator in the GSFC 5 watt VM refrigerator was applied to selection of the core geometry of the preliminary design hot regenerator, described in the next section of this report.

4. Selected Design

The preliminary design hot regenerator is annular in shape with an annulus height of 0.00381m (0.150 in.). The total length is 0.1017m (4 in.), which is packed entirely with 100 mesh stainless steel screen. The screen has a



porosity of 72.5 percent, an area to volume ratio of $10,080\text{m}^2/\text{m}^3$ ($256\text{ in.}^2/\text{in.}^3$) and a hydraulic diameter of 0.000288m (0.01132 in.). The frontal area of the regenerator is 0.000741m^2 (1.15 in.^2).

The screen packing was selected instead of spherical shot, or a combination of the two, in order to minimize pressure drop. The high operating temperature of the hot regenerator minimizes the effect of the increased void volume of screen packing as compared to spheres. Pressure drop through the screen matrix is also predicted more accurately than for the spheres. This is caused by the inherent uniformity of the screen material rather than the analytical methods used. Slight variations in shot size or departure from a perfectly spherical shape may cause deviations in pressure drop. This is an important factor in the hot regenerator packing selection because of the strong influence of pressure drop on drive motor power. Finally, the selection of stainless steel was based on the superior heat sink capacity in the operating temperature range of interest (see Appendix A).

Table 3-8 presents the detailed output from the regenerator analysis computer program for the preliminary design hot regenerator. The output parameters listed in the table are metal temperature, gas temperature, density, pressure, and flow rate. These parameters are printed as functions of time (angular crankshaft displacement) and of position in the regenerator matrix. The angular position is referenced to top dead center of the cold displacer. Node 0 represents the gas inlet face at the sump end, and Node 11 the outlet at the hot end. Nodes 1 through 10 are internal to the matrix. Positive flow rate indicates flow from the sump toward the hot end.

Figures 3-11 through 3-13 present plots of key parameters from the data of Table 3-8. The hot gas and matrix temperatures are plotted as functions of crankshaft position in Figure 3-11. The small difference between the two temperatures is indicative of adequate heat transfer. The matrix temperature swing of 2.6°R shows that the matrix heat capacity is sufficient.

Figure 3-12 indicates a maximum pressure drop of 4480 N/m^2 (0.65 psi). This low value of pressure drop was intentionally designed into the regenerator, in order to assure that the required drive motor power is low.

Any time gas is introduced into the hot end of the machine at a temperature less than the operating level, additional heat must be added to make up for the deficiency. Thus the regenerator inefficiency represents a loss that must be compensated for. This loss may be evaluated by use of Figures 3-11 and 3-13, which presents the gas flow rate at the hot end of the regenerator. Equation 3-1 is used to evaluate the regenerator loss, as was done for the cold regenerator. The cumulative losses are shown as a function of crank angle in Figure 3-14. The total loss per crankshaft revolution is 0.851J (0.0008059 Btu), which translates to a total loss of 5.67 watts at a speed of 400 rpm .



TABLE 3-8

PRELIMINARY DESIGN
HOT REGENERATOR
PERFORMANCE CHARACTERISTICS

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 0^\circ$					
TIME(SEC.) = 4.49994-01			DATE = 19 SEP 72 TIME = 16142:11		
			2301 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.21275+02	6.20000+02	5.53472-01	9.50556+02	7.22990-03
1	6.73518+02	6.71583+02	5.12103-01	9.50537+02	7.11036-03
2	7.57747+02	7.54938+02	4.56951-01	9.50495+02	6.89636-03
3	8.48148+02	8.45195+02	4.09197-01	9.50448+02	6.70414-03
4	9.39601+02	9.36748+02	3.70202-01	9.50396+02	6.52987-03
5	1.03117+03	1.02845+03	3.37072-01	9.50338+02	6.37079-03
6	1.12271+03	1.12012+03	3.08033-01	9.50274+02	6.22475-03
7	1.21422+03	1.21174+03	2.83085-01	9.50203+02	6.08959-03
8	1.30566+03	1.30328+03	2.62228-01	9.50126+02	5.96319-03
9	1.39667+03	1.39435+03	2.45515-01	9.50043+02	5.84337-03
10	1.48363+03	1.48162+03	2.33275-01	9.49953+02	5.72788-03
11	1.53377+03	1.53257+03	2.27839-01	9.49907+02	5.67095-03

REGULAR PRINTOUTS					
$\theta = 30^\circ$					
TIME(SEC.) = 4.62472-01			DATE = 19 SEP 72 TIME = 16142:20		
			2384 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.20740+02	6.20000+02	5.68819-01	9.77836+02	6.40922-03
1	6.72493+02	6.70605+02	5.27095-01	9.77820+02	6.32067-03
2	7.56264+02	7.53437+02	4.70615-01	9.77786+02	6.16201-03
3	8.46607+02	8.43615+02	4.21421-01	9.77747+02	6.01951-03
4	9.38126+02	9.35217+02	3.81194-01	9.77703+02	5.89031-03
5	1.02977+03	1.02698+03	3.47071-01	9.77654+02	5.77238-03
6	1.12134+03	1.11870+03	3.17174-01	9.77600+02	5.66408-03
7	1.21295+03	1.21037+03	2.91505-01	9.77539+02	5.56382-03
8	1.30444+03	1.30197+03	2.70060-01	9.77472+02	5.46997-03
9	1.39544+03	1.39309+03	2.52892-01	9.77400+02	5.38091-03
10	1.48250+03	1.48046+03	2.40333-01	9.77323+02	5.29497-03
11	1.53315+03	1.53187+03	2.34730-01	9.77282+02	5.25257-03

- NOTES: 1. TEMP, $^{\circ}\text{K} = ^{\circ}\text{R}/1.8$
 2. DENSITY, $\text{kg/m}^3 = 16.02 \times \text{LB/FT}^3$
 3. PRESSURE, $\text{N/m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg/sec} = 0.4536 \times \text{LB/SEC}$



TABLE 3-8 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 60^\circ$					
TIME(SEC.) = 4.74920-01			DATE = 19 SEP 72 TIME = 16142127		
			2453 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.20468+02	6.20000+02	5.78603-01	9.95252+02	3.64141-03
1	6.71714+02	6.70183+02	5.36519-01	9.95245+02	3.59861-03
2	7.55110+02	7.52783+02	4.79174-01	9.95229+02	3.52192-03
3	8.45399+02	8.42939+02	4.29088-01	9.95211+02	3.45302-03
4	9.36962+02	9.34569+02	3.88110-01	9.95190+02	3.39058-03
5	1.02466+03	1.02636+03	3.53375-01	9.95167+02	3.33356-03
6	1.12032+03	1.11811+03	3.22953-01	9.95141+02	3.28120-03
7	1.21193+03	1.20980+03	2.96847-01	9.95112+02	3.23270-03
8	1.30345+03	1.30141+03	2.75051-01	9.95080+02	3.18728-03
9	1.39450+03	1.39255+03	2.57617-01	9.95045+02	3.14414-03
10	1.48174+03	1.47996+03	2.44880-01	9.95008+02	3.10248-03
11	1.53264+03	1.53157+03	2.39191-01	9.94988+02	3.08192-03

REGULAR PRINTOUTS					
$\theta = 90^\circ$					
TIME(SEC.) = 4.86999-01			DATE = 19 SEP 72 TIME = 16142131		
			2491 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.20371+02	6.19940+02	5.80090-01	9.97806+02	-2.71775-04
1	6.71419+02	6.70450+02	5.37452-01	9.97821+02	-2.59092-04
2	7.54453+02	7.53053+02	4.80231-01	9.97850+02	-2.36367-04
3	8.44922+02	8.43278+02	4.30032-01	9.97861+02	-2.15955-04
4	9.36497+02	9.34918+02	3.88980-01	9.97867+02	-1.97450-04
5	1.02421+03	1.02669+03	3.54195-01	9.97871+02	-1.80554-04
6	1.11989+03	1.11843+03	3.23727-01	9.97873+02	-1.65035-04
7	1.21150+03	1.21010+03	2.97587-01	9.97876+02	-1.50661-04
8	1.30303+03	1.30168+03	2.75760-01	9.97877+02	-1.37196-04
9	1.39410+03	1.39278+03	2.58324-01	9.97879+02	-1.24407-04
10	1.48137+03	1.48009+03	2.45594-01	9.97880+02	-1.12052-04
11	1.53242+03	1.53172+03	2.39903-01	9.97881+02	-1.05952-04

- NOTES: 1. TEMP, $^\circ K = ^\circ R / 1.8$
 2. DENSITY, $kg/m^3 = 16.02 \times LB/FT^3$
 3. PRESSURE, $N/m^2 = 6.895 \times 10^3 \times PSI$
 4. FLOW, $kg/sec = 0.4536 \times LB/SEC$



TABLE 3-8 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 120^\circ$					
TIME(SEC.) = 4.99983-01			DATE = 19 SEP 72 TIME = 16142136		
			2531 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.20653+02	6.22208+02	5.70403-01	9.84103+02	-4.29506-03
1	6.71802+02	6.74071+02	5.27693-01	9.84112+02	-4.23006-03
2	7.55048+02	7.57440+02	4.71131-01	9.84132+02	-4.11365-03
3	8.45289+02	8.47604+02	4.22162-01	9.84155+02	-4.00901-03
4	9.36838+02	9.39049+02	3.82142-01	9.84181+02	-3.91409-03
5	1.02852+03	1.03064+03	3.48067-01	9.84210+02	-3.82741-03
6	1.12018+03	1.12221+03	3.18230-01	9.84242+02	-3.74778-03
7	1.21178+03	1.21372+03	2.92636-01	9.84278+02	-3.67400-03
8	1.30329+03	1.30513+03	2.71285-01	9.84317+02	-3.60489-03
9	1.39431+03	1.39595+03	2.54246-01	9.84359+02	-3.53924-03
10	1.48151+03	1.48253+03	2.41876-01	9.84405+02	-3.47581-03
11	1.53299+03	1.53500+03	2.36219-01	9.84428+02	-3.44451-03

REGULAR PRINTOUTS					
$\theta = 150^\circ$					
TIME(SEC.) = 5.12418-01			DATE = 19 SEP 72 TIME = 16142143		
			2601 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.21459+02	6.23311+02	5.54740-01	9.57837+02	-6.61087-03
1	6.72930+02	6.75554+02	5.12932-01	9.57853+02	-6.49170-03
2	7.56209+02	7.58960+02	4.58015-01	9.57890+02	-6.27824-03
3	8.46384+02	8.49030+02	4.10516-01	9.57930+02	-6.08637-03
4	9.37860+02	9.40373+02	3.71690-01	9.57976+02	-5.91229-03
5	1.02949+03	1.03187+03	3.38576-01	9.58026+02	-5.75330-03
6	1.12109+03	1.12336+03	3.09564-01	9.58082+02	-5.60727-03
7	1.21263+03	1.21479+03	2.84663-01	9.58144+02	-5.47205-03
8	1.30409+03	1.30611+03	2.63877-01	9.58210+02	-5.34546-03
9	1.39500+03	1.39677+03	2.47282-01	9.58282+02	-5.22535-03
10	1.48195+03	1.48306+03	2.35225-01	9.58359+02	-5.10943-03
11	1.53370+03	1.53500+03	2.29724-01	9.58398+02	-5.05227-03

- NOTES: 1. TEMP, $^{\circ}\text{K} = ^{\circ}\text{R}/1.8$
 2. DENSITY, $\text{kg}/\text{m}^3 = 16.02 \times \text{LB}/\text{FT}^3$
 3. PRESSURE, $\text{N}/\text{m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg}/\text{sec} = 0.4536 \times \text{LB}/\text{SEC}$



TABLE 3-8 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 180^\circ$			DATE = 19 SEP 72 TIME = 16145153		
TIME(SEC.) = 5.24999-01			2686 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.22530+02	6.24527+02	5.37248-01	9.28427+02	-7.03001-03
1	6.74434+02	6.77269+02	4.96408-01	9.28446+02	-6.91612-03
2	7.57757+02	7.60724+02	4.43322-01	9.28487+02	-6.71210-03
3	8.47853+02	8.50717+02	3.97448-01	9.28533+02	-6.52868-03
4	9.39242+02	9.41973+02	3.59942-01	9.28585+02	-6.36224-03
5	1.03079+03	1.03340+03	3.27892-01	9.28642+02	-6.21025-03
6	1.12233+03	1.12482+03	2.99800-01	9.28706+02	-6.07068-03
7	1.21382+03	1.21620+03	2.75672-01	9.28776+02	-5.94151-03
8	1.30520+03	1.30745+03	2.55521-01	9.28853+02	-5.82069-03
9	1.39598+03	1.39795+03	2.39428-01	9.28936+02	-5.70617-03
10	1.48255+03	1.48377+03	2.27751-01	9.29024+02	-5.59580-03
11	1.53422+03	1.53500+03	2.22441-01	9.29070+02	-5.54142-03

REGULAR PRINTOUTS					
$\theta = 210^\circ$			DATE = 19 SEP 72 TIME = 16143102		
TIME(SEC.) = 5.37500-01			2770 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.23545+02	6.25378+02	5.22578-01	9.03483+02	-5.58501-03
1	6.75859+02	6.78443+02	4.82627-01	9.03496+02	-5.49444-03
2	7.59223+02	7.61920+02	4.31048-01	9.03527+02	-5.33221-03
3	8.49249+02	8.51850+02	3.86504-01	9.03562+02	-5.18636-03
4	9.40560+02	9.43040+02	3.50080-01	9.03601+02	-5.05401-03
5	1.03204+03	1.03441+03	3.18911-01	9.03645+02	-4.93315-03
6	1.12352+03	1.12578+03	2.91579-01	9.03693+02	-4.82219-03
7	1.21496+03	1.21711+03	2.68092-01	9.03746+02	-4.71954-03
8	1.30628+03	1.30831+03	2.48464-01	9.03805+02	-4.62361-03
9	1.39693+03	1.39870+03	2.32782-01	9.03868+02	-4.53277-03
10	1.48313+03	1.48422+03	2.21404-01	9.03935+02	-4.44531-03
11	1.53452+03	1.53500+03	2.16237-01	9.03970+02	-4.40226-03

- NOTES: 1. TEMP, $^\circ K = ^\circ R / 1.8$
 2. DENSITY, $kg/m^3 = 16.02 \times LB/FT^3$
 3. PRESSURE, $N/m^2 = 6.895 \times 10^3 \times PSIA$
 4. FLOW, $kg/sec = 0.4536 \times LB/SEC$



TABLE 3-8 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 240^\circ$					
TIME(SEC.) = 5.49883-01			DATE = 19 SEP 72 TIME = 16143110		
			2839 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.24255+02	6.25640+02	5.13767-01	8.88139+02	-3.01700-03
1	6.76861+02	6.78778+02	4.74427-01	8.88145+02	-2.96216-03
2	7.60253+02	7.62243+02	4.23728-01	8.88159+02	-2.86393-03
3	8.50233+02	8.52136+02	3.79949-01	8.88174+02	-2.77560-03
4	9.41492+02	9.43292+02	3.44146-01	8.88192+02	-2.69545-03
5	1.03293+03	1.03464+03	3.13495-01	8.88211+02	-2.62226-03
6	1.12437+03	1.12599+03	2.86607-01	8.88232+02	-2.55508-03
7	1.21577+03	1.21731+03	2.63492-01	8.88256+02	-2.49296-03
8	1.30705+03	1.30849+03	2.44165-01	8.88282+02	-2.43493-03
9	1.39761+03	1.39885+03	2.28714-01	8.88310+02	-2.38002-03
10	1.48355+03	1.48431+03	2.17491-01	8.88340+02	-2.32720-03
11	1.53467+03	1.53500+03	2.12391-01	8.88356+02	-2.30121-03

REGULAR PRINTOUTS					
$\theta = 270^\circ$					
TIME(SEC.) = 5.62294-01			DATE = 19 SEP 72 TIME = 16143114		
			2879 = COUNT (NO. OF CALCULATIONS)		
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.24534+02	6.20000+02	5.17611-01	8.87017+02	1.80261-04
1	6.77266+02	6.79166+02	4.73571-01	8.87013+02	1.48452-04
2	7.60675+02	7.62631+02	4.22981-01	8.87008+02	9.14750-05
3	8.50643+02	8.52644+02	3.79239-01	8.87008+02	4.02592-05
4	9.41867+02	9.43930+02	3.43470-01	8.87008+02	-6.21042-06
5	1.03332+03	1.03539+03	3.12841-01	8.87008+02	-4.86358-05
6	1.12474+03	1.12684+03	2.85986-01	8.87008+02	-8.75760-05
7	1.21614+03	1.21824+03	2.62909-01	8.87008+02	-1.23582-04
8	1.30741+03	1.30950+03	2.43623-01	8.87009+02	-1.57213-04
9	1.39793+03	1.39995+03	2.28211-01	8.87010+02	-1.89036-04
10	1.48374+03	1.48481+03	2.17104-01	8.87012+02	-2.19656-04
11	1.53474+03	1.53500+03	2.12061-01	8.87013+02	-2.34723-04

- NOTES: 1. TEMP, $^\circ\text{K} = ^\circ\text{R}/1.8$
 2. DENSITY, $\text{kg/m}^3 = 16.02 \times \text{LB/FT}^3$
 3. PRESSURE, $\text{N/m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg/sec} = 0.4536 \times \text{LB/SEC}$



TABLE 3-8 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 300^\circ$					
TIME(SEC.) = 5.74980-01					
DATE = 19 SEP 72 TIME = 16143118					
2916 = COUNT (NO. OF CALCULATIONS)					
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.23531+02	6.20000+02	5.24427-01	8.99072+02	3.45505-03
1	6.77067+02	6.75680+02	4.82270-01	8.99064+02	3.38276-03
2	7.60439+02	7.58704+02	4.30755-01	8.99048+02	3.25325-03
3	8.50397+02	8.48574+02	3.86067-01	8.99029+02	3.13686-03
4	9.41660+02	9.39914+02	3.49467-01	8.99008+02	3.03128-03
5	1.03311+03	1.03146+03	3.18239-01	8.98986+02	2.93492-03
6	1.12455+03	1.12300+03	2.90846-01	8.98961+02	2.84648-03
7	1.21597+03	1.21450+03	2.67292-01	8.98934+02	2.76471-03
8	1.30725+03	1.30588+03	2.47590-01	8.98904+02	2.68834-03
9	1.39780+03	1.39652+03	2.31815-01	8.98872+02	2.61608-03
10	1.48362+03	1.48254+03	2.20318-01	8.98837+02	2.54657-03
11	1.53465+03	1.53396+03	2.15061-01	8.98819+02	2.51236-03

REGULAR PRINTOUTS					
$\theta = 330^\circ$					
TIME(SEC.) = 5.87449-01					
DATE = 19 SEP 72 TIME = 16143126					
2985 = COUNT (NO. OF CALCULATIONS)					
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	6.22275+02	6.20000+02	5.37147-01	9.21596+02	5.99175-03
1	6.76343+02	6.74470+02	4.94867-01	9.21582+02	5.88841-03
2	7.59518+02	7.57030+02	4.42205-01	9.21548+02	5.70319-03
3	8.49443+02	8.46809+02	3.96298-01	9.21511+02	5.53672-03
4	9.40752+02	9.38198+02	3.58643-01	9.21469+02	5.38577-03
5	1.03226+03	1.02982+03	3.26566-01	9.21423+02	5.24796-03
6	1.12375+03	1.12143+03	2.98434-01	9.21371+02	5.12149-03
7	1.21520+03	1.21299+03	2.74252-01	9.21314+02	5.00450-03
8	1.30653+03	1.30442+03	2.54031-01	9.21253+02	4.89517-03
9	1.39712+03	1.39514+03	2.37846-01	9.21186+02	4.79164-03
10	1.48304+03	1.48130+03	2.26050-01	9.21115+02	4.69196-03
11	1.53429+03	1.53319+03	2.20628-01	9.21077+02	4.64289-03

- NOTES: 1. TEMP, $^\circ K = ^\circ R / 1.8$
 2. DENSITY, $kg/m^3 = 16.02 \times LB/FT^3$
 3. PRESSURE, $N/m^2 = 6.895 \times 10^3 \times PSI$
 4. FLOW, $kg/sec = 0.4536 \times LB/SEC$

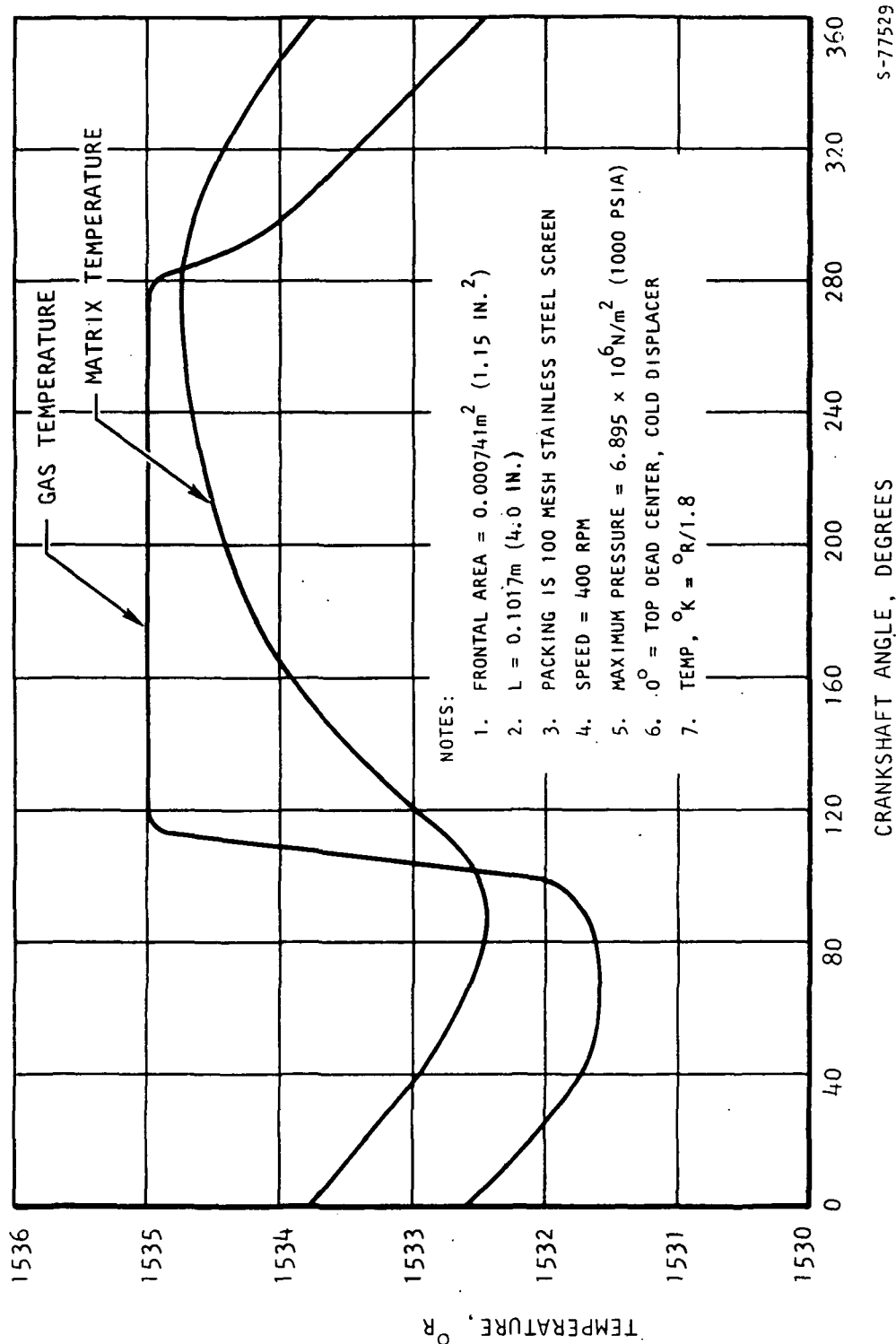


TABLE 3-8 (Continued)

NODE NO.	MATRIX TEMPERATURE	GAS TEMPERATURE	GAS DENSITY	GAS PRESSURE	GAS FLOW RATE
REGULAR PRINTOUTS					
$\theta = 360^\circ$ TIME(SEC.) = 5.99999-01 DATE = 19 SEP 72 TIME = 16:43:35 3068 = COUNT (NO. OF CALCULATIONS)					
N	TM(N) DEG.R	TG(N) DEG.R	RG(N) LBM/CF	PG(N) PSIA	WG(N) LBM/SEC
-0	5.21331+02	6.20000+02	5.53471-01	9.50554+02	7.22987-03
1	6.75309+02	6.73293+02	5.10820-01	9.50535+02	7.11062-03
2	7.58153+02	7.55403+02	4.56675-01	9.50493+02	6.89674-03
3	8.48017+02	8.45090+02	4.09247-01	9.50447+02	6.70450-03
4	9.39386+02	9.36538+02	3.70282-01	9.50395+02	6.53019-03
5	1.03096+03	1.02824+03	3.37141-01	9.50337+02	6.37106-03
6	1.12257+03	1.11993+03	3.08088-01	9.50273+02	6.22499-03
7	1.21403+03	1.21155+03	2.83132-01	9.50202+02	6.08980-03
8	1.30542+03	1.30305+03	2.62276-01	9.50125+02	5.96337-03
9	1.39601+03	1.39384+03	2.45597-01	9.50041+02	5.84350-03
10	1.48213+03	1.48015+03	2.33449-01	9.49952+02	5.72794-03
11	1.53371+03	1.53246+03	2.27848-01	9.49906+02	5.67101-03

- NOTES: 1. TEMP, $^{\circ}\text{K} = ^{\circ}\text{R}/1.8$
 2. DENSITY, $\text{kg}/\text{m}^3 = 16.02 \times \text{LB}/\text{FT}^3$
 3. PRESSURE, $\text{N}/\text{m}^2 = 6.895 \times 10^3 \times \text{PSI}$
 4. FLOW, $\text{kg}/\text{sec} = 0.4536 \times \text{LB}/\text{SEC}$





S-77529

Figure 3-11. Temperature Variation at Hot End of Preliminary Design Hot Regenerator



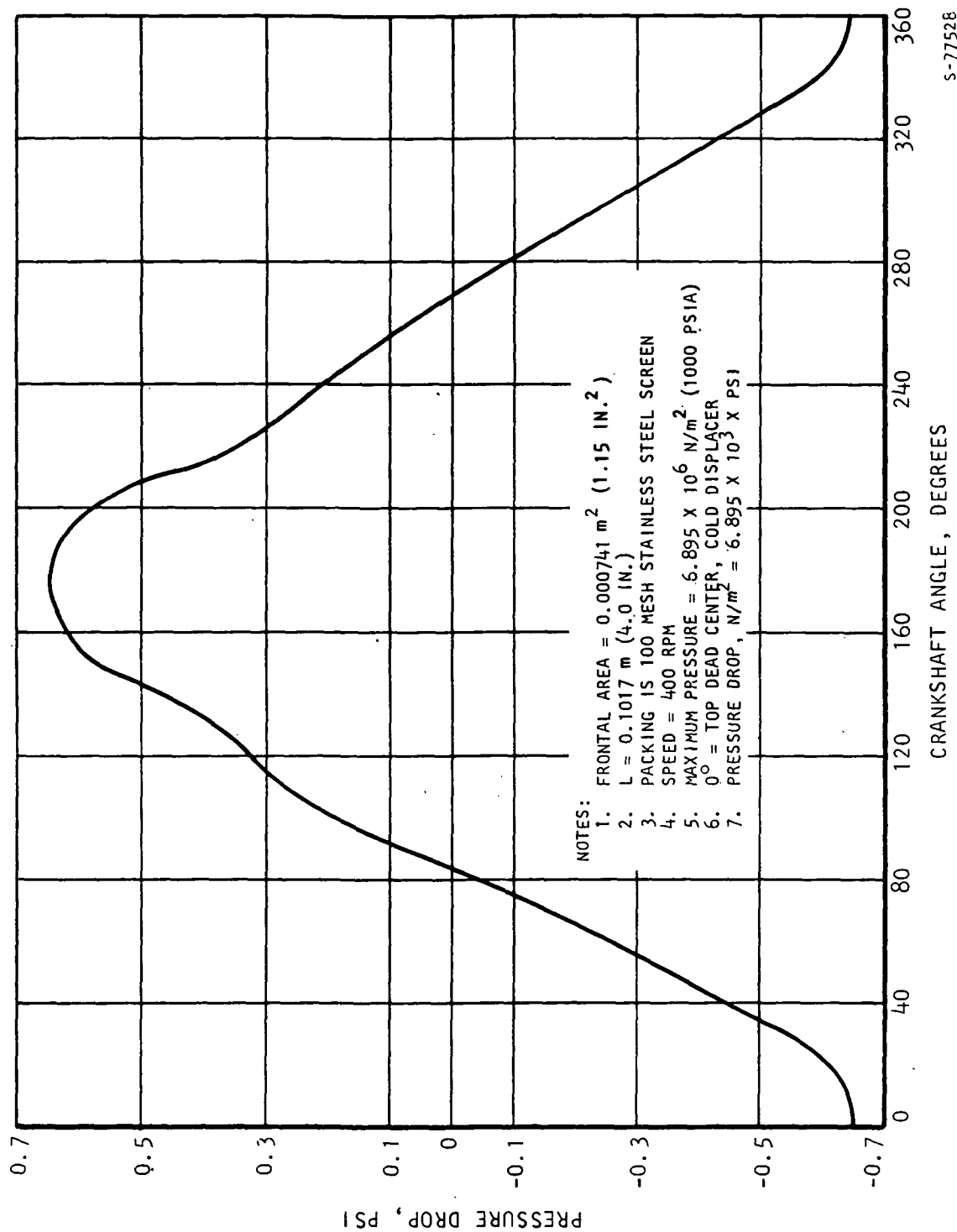
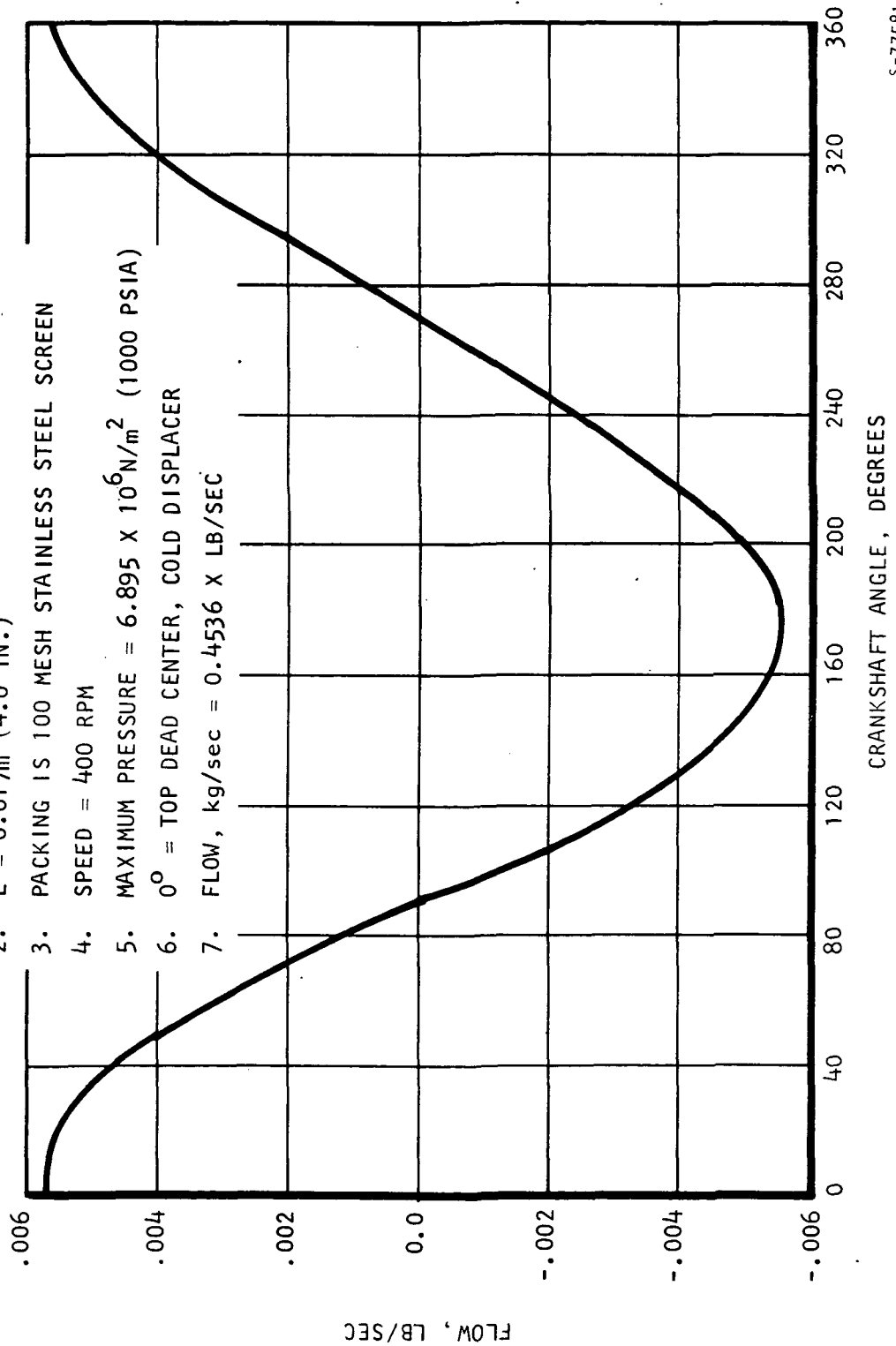


Figure 3-12. Pressure Drop of Preliminary Design Hot Regenerator



NOTES:

1. FRONTAL AREA = 0.000741 m^2 (1.15 IN.^2)
2. $L' = 0.017 \text{ m}$ (4.0 IN.)
3. PACKING IS 100 MESH STAINLESS STEEL SCREEN
4. SPEED = 400 RPM
5. MAXIMUM PRESSURE = $6.895 \times 10^6 \text{ N/m}^2$ (1000 PSIA)
6. 0° = TOP DEAD CENTER, COLD DISPLACER
7. FLOW, $\text{kg/sec} = 0.4536 \times \text{LB/SEC}$



S-77581

Figure 3-13. Hot End Flowrate of Preliminary Design Hot Regenerator

NOTES:

1. LOSS = $\int_0^{360} W_C (T - T_{REF}) d\theta$
2. FRONTAL AREA = $0.000741 m^2 (1.15 \text{ IN.})$
3. $L = 0.1017 m (4.0 \text{ IN.})$
4. PACKING IS 100 MESH STAINLESS STEEL SCREEN
5. SPEED = 400 RPM
6. MAXIMUM PRESSURE = $6.895 \times 10^6 N/m^2 (1000 \text{ PSIA})$
7. 0° = TOP DEAD CENTER, COLD DISPLACER
8. LOSS, JOULES = $1055 \times \text{BTU}$

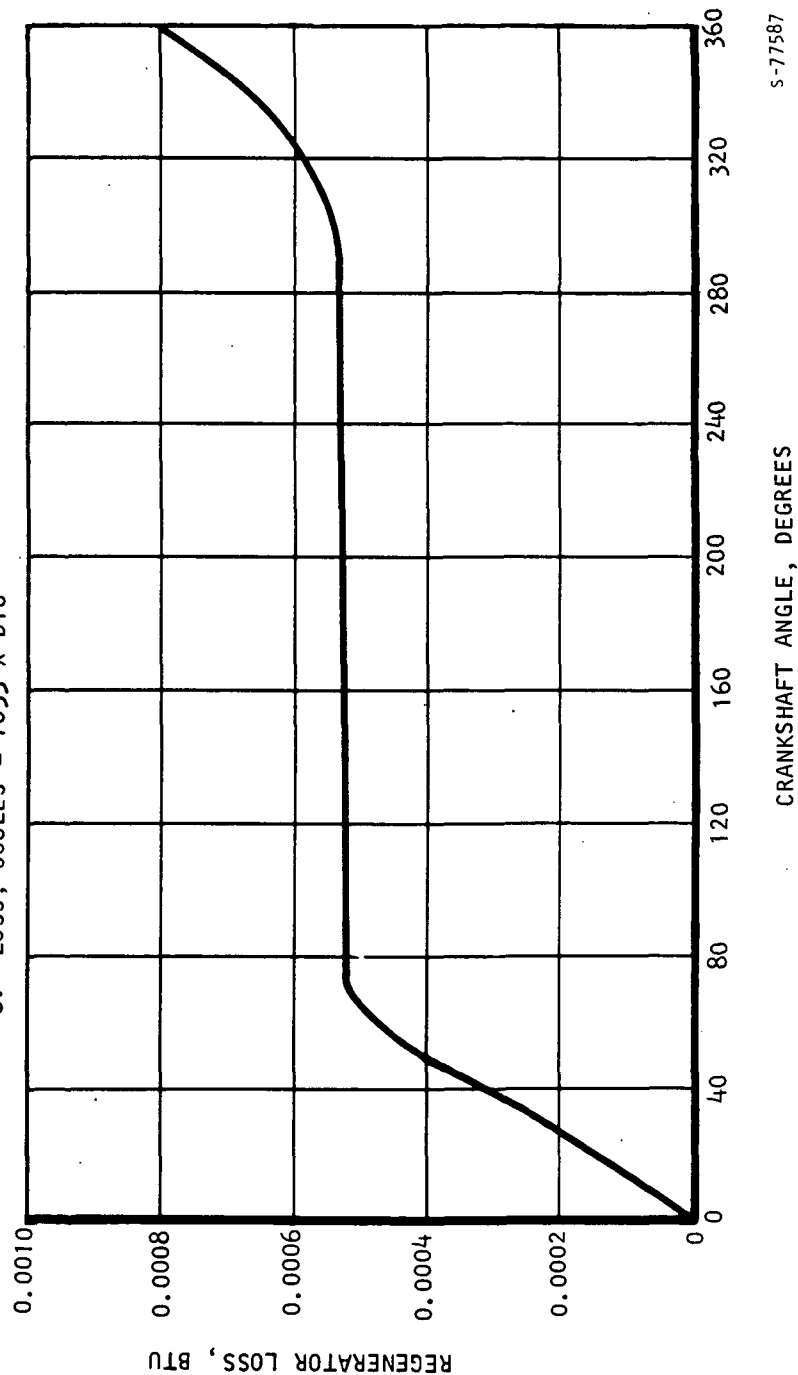


Figure 3-14. Preliminary Design Hot Regenerator Thermal Loss per Cycle



Cold End Heat Exchanger Design

1. Introduction

The cold end heat exchanger transfers the refrigeration heat load from the cold finger of the VM refrigerator to the working fluid. The primary design criteria are:

Minimum temperature difference between working fluid and refrigeration load--This is an extremely important consideration. Since the refrigeration temperature is fixed, the larger the temperature difference, the lower the working fluid temperature the refrigerator must produce with inherent decreased thermodynamic efficiencies.

Low working fluid pressure drop--The heat exchanger must provide high thermal performance and still not impose excessive working fluid pressure drop on the system. The heat exchanger pressure drop subtracts directly from the cold expansion volume pressure variations, which produce the refrigeration.

Low void or internal volume--Void volume at low temperature significantly decreases the refrigeration capacity of VM refrigerators. An evaluation of this effect is presented elsewhere in this report.

Flow distribution--Uniform flow within each element of the heat exchanger must occur. Potential problems here are twofold: nonuniform flow leads to (1) reduced heat exchanger conductance, and (2) fluid elements at different temperatures. Subsequent mixing of these elements results in entropy increase and reduces the thermodynamic efficiency of the refrigerator.

Heat exchanger interfaces--The cold end heat exchanger interfaces with the cold regenerator, the cold expansion volume, and the refrigeration heat load. These interface requirements control the heat exchanger configuration to some extent. The annular flow passages of the heat exchanger interface with the cold regenerator through a perforated plate flow distributor and with the cold expansion volume through ports in the displaced volume dome. The interface with the refrigeration heat load is provided by means of an axial clamp and a high thermal conductivity (copper plating) heat path to the heat exchanger outer wall.

2. Design Iterations

The heat exchanger was initially configured in a similar manner as that of the GSFC 5 watt VM. The cold finger of the machine was essentially a constant diameter for the entire length. At the interface between the cold regenerator and the cold heat exchanger, the flow was directed radially outward to the heat exchanger slots. After traversing the slots in an axial direction, the gas was turned radially inward, into a constant area passage, and then entered the cold expansion volume.



After several design iterations, it was found that a heat exchanger configuration with twelve rectangular slots 0.00953m (0.375 in.) long, and 0.001017 (0.04) x 0.002034m (0.08 in.) in cross section provided the desired design characteristics. This heat exchanger required the use of a relatively large filler block between the regenerator inner wall and the outer pressure vessel wall. The cold end of the refrigerator was subsequently changed because of stress considerations. The heat exchanger was reconfigured, but the essential design features remained the same. The selected design, which is the direct result of the cold finger reconfiguration, is discussed below.

3. Selected Design

The preliminary design cold heat exchanger is shown in Figure 3-15. Twelve axial slots are arranged in an annular configuration around the regenerator inner wall. This configuration, with the reduction in outer wall diameter at the cold end of the regenerator, eliminated the use of a relatively heavy filler block for the heat exchanger, and thus reduced the cantilevered weight of the cold end.

The cyclic flow enters (and exits) the heat exchanger from the cold expansion volume via ports at the outer diameter of the displacer bore. The working fluid traverses the axial slots and enters the radial distribution channels. The flow from the channels is distributed uniformly over the cold face of the regenerator by the perforated plate flow distributor.

The twelve flow channels are 0.00953m (0.375 in.) long, and are rectangular in shape. The cross section of each channel is 0.001017 (0.04) x 0.002034m (0.08 in.), with the shorter side in the radial direction. This provides a minimum radial height and a maximum prime surface for heat transfer.

The average flow during the cyclic flow process is used for heat transfer performance evaluation. The total heat transfer conductance (ηhA) of the cold heat exchanger is 0.403 watts/ $^{\circ}$ K (0.224 watts/ $^{\circ}$ R). At a heat load of 0.25 watts, the corresponding film temperature drop from the metal heat exchanger wall to the working fluid is 0.621 $^{\circ}$ K (1.12 $^{\circ}$ R). The pressure drop, evaluated at the maximum flow rate of the cycle, is 94.5 N/m² (0.0137 psi).

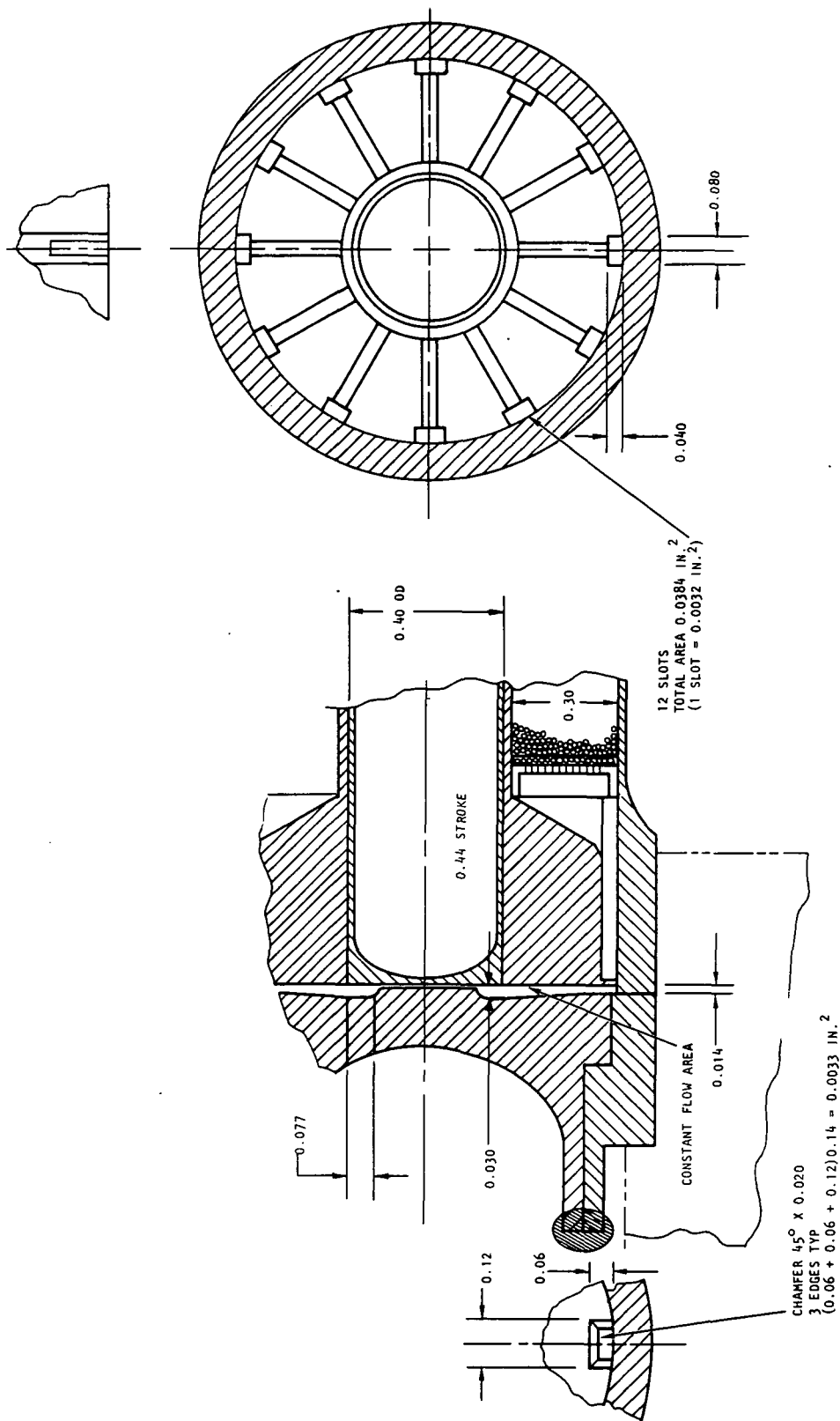
The refrigeration heat load is mounted on a removable copper end cap. This end cap is clamped to the cold end of the machine with spring loaded axial bolts, and indium foil is employed at the interface. The outer diameter of the cold end of the refrigerator is copper plated, in order to allow uniform heat distribution over the outer surface of the heat exchanger. Thus a minimum temperature drop from the refrigeration load to the heat exchanger wall is assured.

Hot End Heat Exchanger Design

1. Introduction

The hot-end heat exchanger transfers thermal energy to the working fluid of the VM refrigerator, thus providing the power input necessary to drive the system. Critical design criteria are similar to those of the cold heat exchanger, with emphasis on various parameters changed.





S-77573 -A

Figure 15. Schematic of Cold End Heat Exchanger

Low working fluid pressure drop--As with the hot regenerator, low pressure drop is of prime importance in the hot end heat exchanger. This parameter directly affects drive motor power, and thus must be minimized. In addition, pressure drop reduces the cold end pressure fluctuations, and thus refrigeration capacity. The high fluid temperature, and thus low density, in the hot end heat exchanger requires that careful attention be paid to pressure drop. High fluid velocities occur for flow rates and cross-sectional areas similar to lower temperature portions of the machine.

Low Film Temperature Drop--The maximum temperature of the heat exchanger wall is set by structural considerations. Thus a low film temperature drop allows a close working fluid approach to this maximum temperature. This in turn increases thermodynamic efficiency, which is a direct function of hot end gas temperature.

Flow distribution--Non-uniform flow in the hot end heat exchanger must be avoided for the same reasons given for the cold end heat exchanger.

Low void or internal volume--The hot end of the VM refrigerator is the least sensitive portion of the machine to void volume. This dependency, caused by the high operating temperature, is discussed elsewhere in this report.

Heat exchanger interfaces--The hot heat exchanger provides working fluid transitions with both the hot displaced volume and the hot regenerator. The electrical heater, which provides the energy input to the system is bonded directly to the outer surface of the heat exchanger--pressure dome. The heat input is transferred directly to the working fluid from the heat exchanger walls.

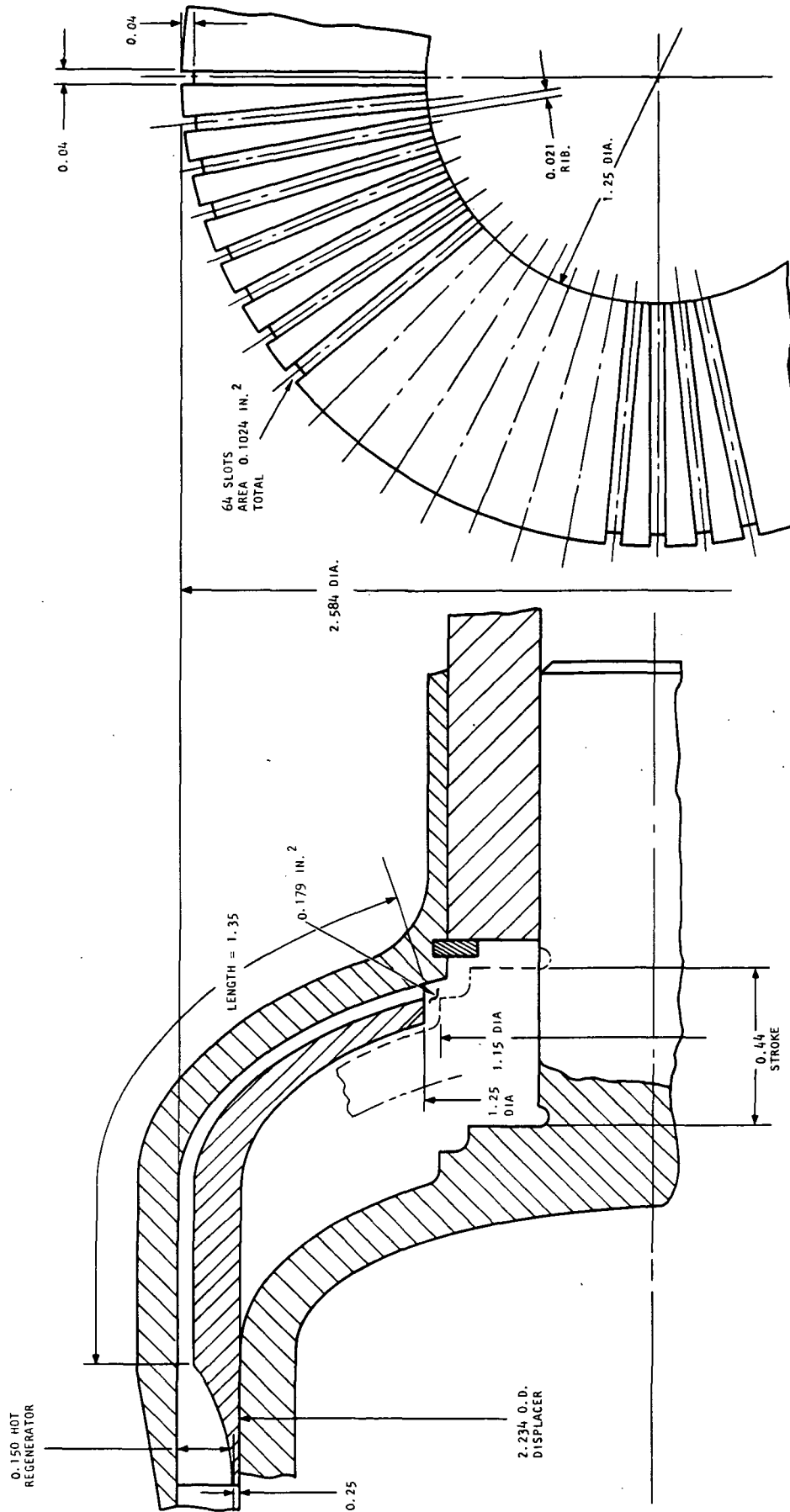
2. Design Configuration

The hot end heat exchanger was patterned after that used on the GSFC 5 watt VM, with modifications that simplify the required manufacturing processes. The preliminary design configuration is shown in Figure 3-16.

The working fluid enters and exits the heat exchanger at the hot displaced volume interface by termination of the heat exchanger inner wall at a diameter corresponding to the hot end bearing support. The flow enters a total of 64 square passages contained between the heat exchanger inner wall and the outer pressure dome. These passages direct the flow initially in a radially outward direction. The passages follow the pressure dome contour, until the flow emerges in an axial direction into transition slots that form the interface with the hot regenerator.

The flow passages extend the full length of the heat exchanger, with no changes in cross-sectional area or number of slots. This configuration provides adequate heat transfer and a very low pressure drop, which are the desired characteristics of the heat exchanger. The constant cross-sectional area slots are simpler to produce in manufacture than intersecting ribs and variable flow cross-sectional area.





S-77553-A

Figure 3-16. Schematic of Hot End Heat Exchanger



Heat transfer to the working fluid is accomplished at both primary and secondary heat transfer surfaces. The primary surface consists of the portions of the outer pressure dome which form the boundaries of the slots. Heat is transferred directly to the working fluid from this surface. The secondary heat transfer surface is composed of the sides and inner walls of the flow passages. Heat is conducted from the other surface to these secondary surfaces, and then to the working fluid. The secondary surface is approximately 93 percent as effective as the primary surface.

The heat transfer performance calculation is based on the average value of working fluid flow rate during the cycle, which is considered a conservative estimate. The flow passage length is less than 40 hydraulic diameters, which is generally too short for fully developed laminar flow to occur. However, under the further conservative assumption that fully developed laminar flow does occur, the total $\eta h A$ of the hot heat exchanger is 6.51 watts/ $^{\circ}\text{K}$ (3.62 watts/ $^{\circ}\text{R}$). This translates to a film temperature drop of 12.28 $^{\circ}\text{K}$ (22.1 $^{\circ}\text{R}$) at a total heat load of 80 watts.

The pressure drop of the hot heat exchanger is calculated at the maximum working fluid flow rate during the cycle, and thus represents worst case conditions. The pressure drop under these conditions is 1179 N/m^2 (0.171 psi), which is considered an acceptable value.

Ambient Sump Heat Exchanger

1. Introduction

The sump heat exchanger functions to transfer heat from the working fluid of the VM refrigerator for rejection from the system. The design criteria for this heat exchanger are similar to those of the cold- and hot-end heat exchangers with changed emphasis on the various items. The preliminary design criteria consist of:

- Low Working Fluid Pressure Drop--The heat exchanger must provide good thermal performance and yet not lead to an excessive pressure drop of the working fluid. As with the cold and hot end heat exchangers; the pressure drop subtracts from the pressure-volume variations in the cold expansion volume thereby reducing the refrigeration capacity. In addition, the pressure drop across the sump heat exchanger can have a significant affect on the drive motor power requirements unless this pressure drop is minimized.
- Low Void or Internal Volume--Void volumes reduce the refrigeration capacity of VM refrigerators by decreasing the pressure variations or pressure ratio; minimization of the heat exchanger internal void volume is therefore important. The relative importance of void volume in this heat exchanger as compared to other regions of the machine is discussed elsewhere in this report.



- Minimization of the Film Temperature Drop--The thermodynamic efficiency of the refrigerator increases as the temperature of the gas in the sump decreases. Minimization of the sump heat exchanger film temperature drop allows maximum performance for the fixed heat rejection and heat sink temperature.
- Flow Distribution--Uniform flow within the heat exchanger is important for two reasons: (1) non-uniform flow leads to reduced conductance of the heat exchanger and (2) non-uniform flow leads to fluid elements at different temperatures; subsequent mixing of these elements results in an increase in entropy and reduced thermodynamic efficiency of the refrigerator.
- Heat Exchanger Interfaces--The sump heat exchanger must interface with both the hot and cold regenerators, fluid passages into the sump volume, and a cooling collar or clamp which provides the heat sink.

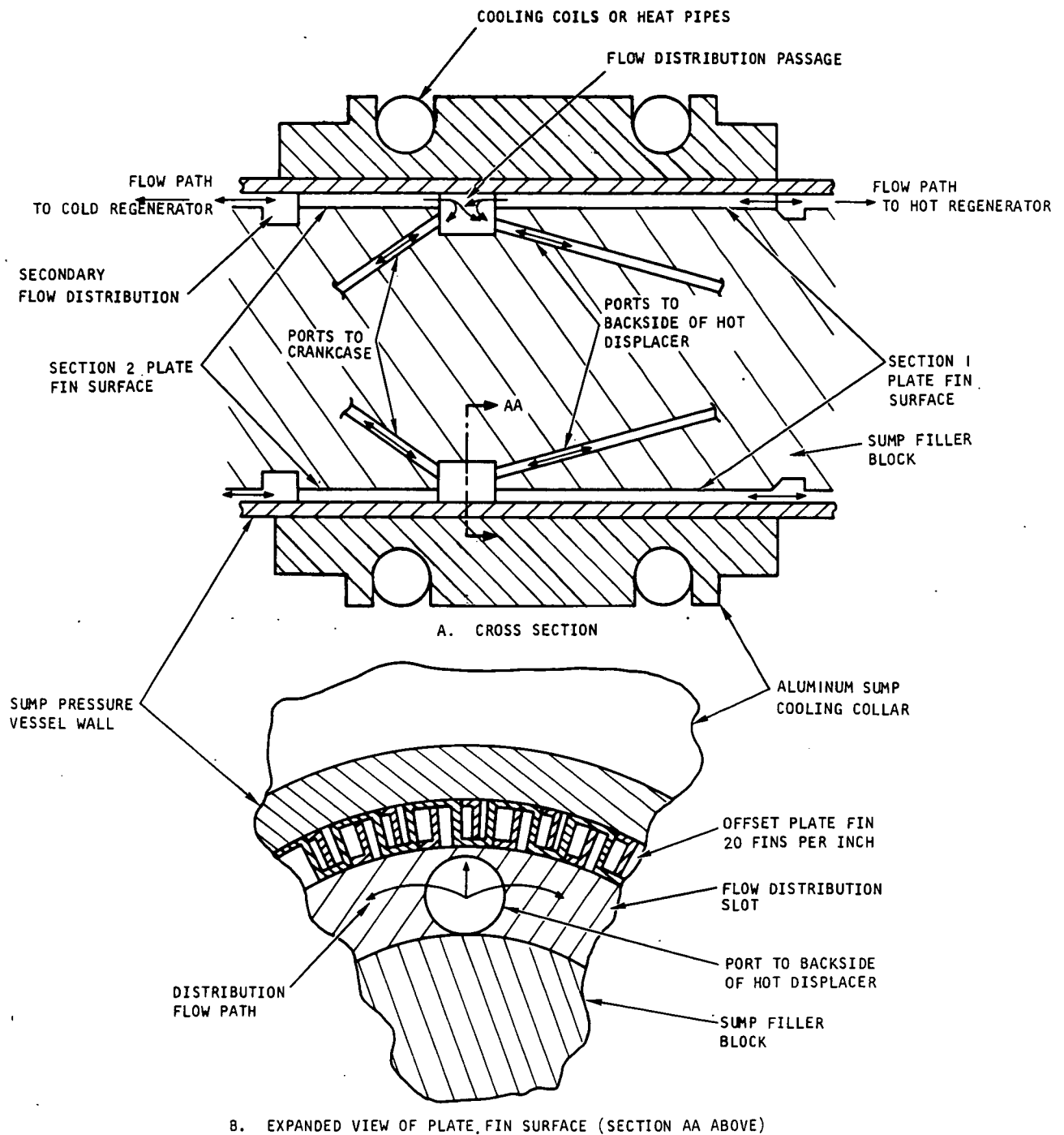
2. Design Configuration

The configuration of the ambient sump heat exchanger is shown in Figure 3-17. This configuration is a refinement of the design evolved for the GSFC 5 watt VM.

The annular shaped heat exchanger is divided into two sections as shown, with both sections being identical in configuration except for length. The heat transfer surface of each section is formed by brazing an offset copper plate fin to the inside surface of the cylindrical section of the sump pressure vessel wall. The cylindrical sump filler block fits inside the plate fin thereby forming an annular passage forcing flow through the finned surface. At the right hand side (Figure 3-17), flow of the working fluid enters and exits Section 1 of the heat exchanger as it flows to and from the hot regenerator during the cyclic flow process. The average flow rate in this section of the heat exchanger is approximately 3 times that in Section 2; this accounts for the greater length (larger heat transfer surface) required for this section. At the other end of Section 1 of the heat exchanger, the flow enters and exits from a flow distribution passage cut into the sump filler block. This distribution passage or slot is supplied working fluid via ports that connect to the active cycle volumes in the crank case and behind the hot displacer as shown. The distribution slot is sized to provide uniform flow across the face of the heat exchanger.

On the left hand side of Figure 3-17 flow enters and exits Section 2 of the heat exchanger as it flows to and from the cold regenerator. This section of the heat exchanger is pneumatically connected to the cold regenerator via the sump filler block (not shown in Figure 3-17) and the cold-end linear bearing support. The right hand end of this section of the heat exchanger interfaces and shares the central flow distribution slot with the other section of the heat exchanger.





S-73484

Figure 3-17. Sump Heat Exchanger Configuration



The path for heat transfer from both sections of the heat exchanger is from the gas to the plate fin surface, from the finned surface through the pressure vessel wall and on into the aluminum sump cooling collar. Indium foil is placed between the sump pressure vessel and the cooling collar; this foil is maintained under a 100 psi interface pressure to ensure good thermal contact. Heat is finally rejected from the system to water cooling coils brazed into channels cut in the cooling collar. This collar is designed to allow interfacing of the refrigerator with water cooling coils or ammonia heat pipes interchangeably.

3. Heat Exchanger Characterization

The rate of heat transfer for each section of sump heat exchanger can be expressed as:

$$\dot{Q} = h(A_p + \eta_f A_f) \overline{\Delta T} \quad (3-2)$$

where h = the average heat transfer coefficient .

A_p = basic area of the plate

η_f = fin effectiveness

A_f = fin area

$\overline{\Delta T}$ = average temperature difference between the working fluid and the heat transfer surface

Referring to Figure 3-18 the following relations can be derived:

Plate area

$$A_p = (1 - N\delta)WL \quad (3-3)$$

where N = fins per inch

W = plate width

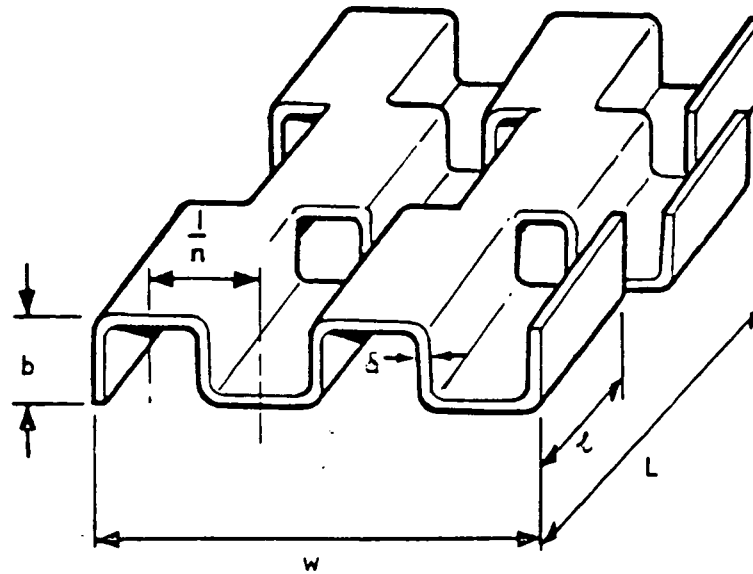
L = plate length

Fin area

$$A_f = \left\{ 2N(b - \delta) + \frac{N}{2} \left(\frac{1}{N} - \delta \right) \right\} WL \quad (3-4)$$

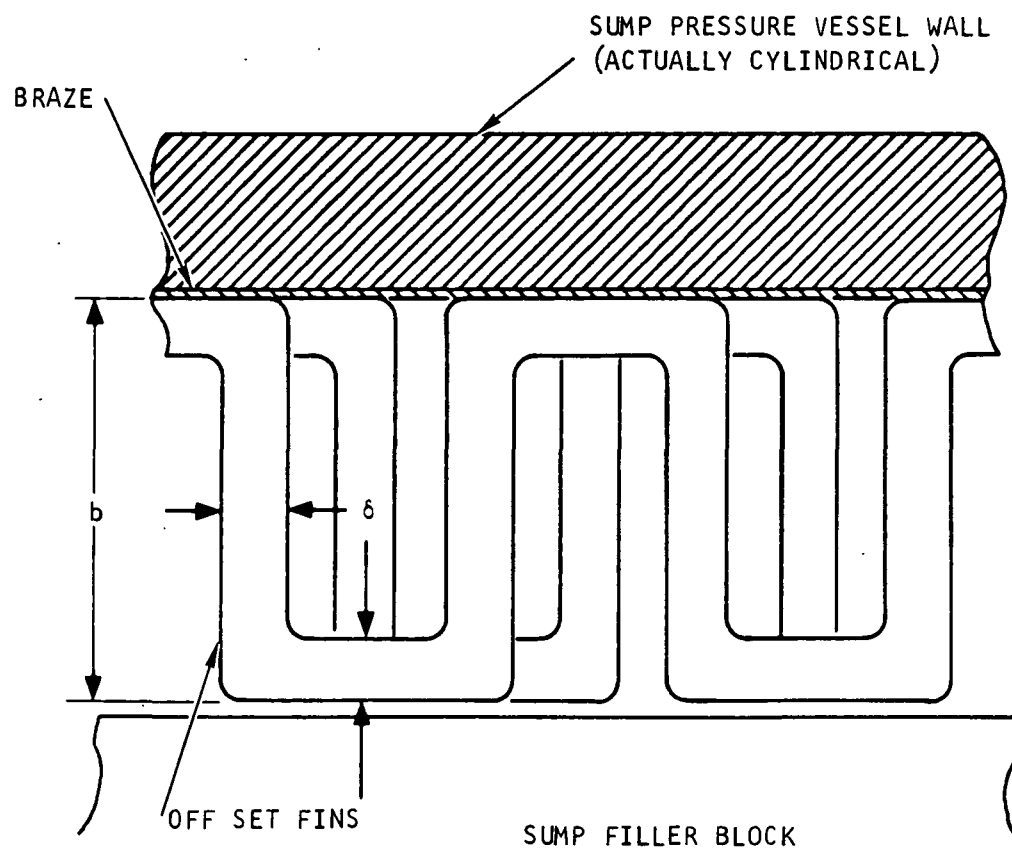
Note: This neglects the fin area exposed between fin and the sump filler block--a conservative approach.





a. NOMENCLATURE FOR OFFSET FIN

S-41922



b. INSTALLED PLATE-FIN SURFACE

S-73483

Figure 3-18. Rectangular Offset Plate-Fin VM Refrigerator Sump Heat Exchanger



Fin effectiveness

$$\eta_f = \frac{\text{Tanh } (ML_e)}{ML_e} \quad (3-5)$$

$$\text{where } M = \sqrt{\frac{2h}{k\delta}} \quad (3-6)$$

k = fin material thermal conductivity and the fin length L_e is given by

$$L_e = b + \frac{1}{2} \left(\frac{1}{N} - \delta \right) \quad (3-7)$$

Flow cross sectional area

$$A_f = \left\{ b - \delta (N(b - \delta) + 1) \right\} W \quad (3-8)$$

Hydraulic diameter

$$D_H = \frac{2(b - \delta) \left(\frac{1}{N} - \delta \right)}{(b - \delta) + \left(\frac{1}{N} - \delta \right)} \quad (3-9)$$

Performance characteristics unique to a given plate fin surface are generally presented as plots of Colburn's j factor and Fanning's friction factor f as functions of Reynolds Number. Colburn's j factor is defined by:

$$j = \frac{h}{C_p G} Pr^{2/3} \quad (3-10)$$

where C_p = gas heat capacity

Pr = Prandtl number

$$G = \frac{\dot{m}}{A_f}$$

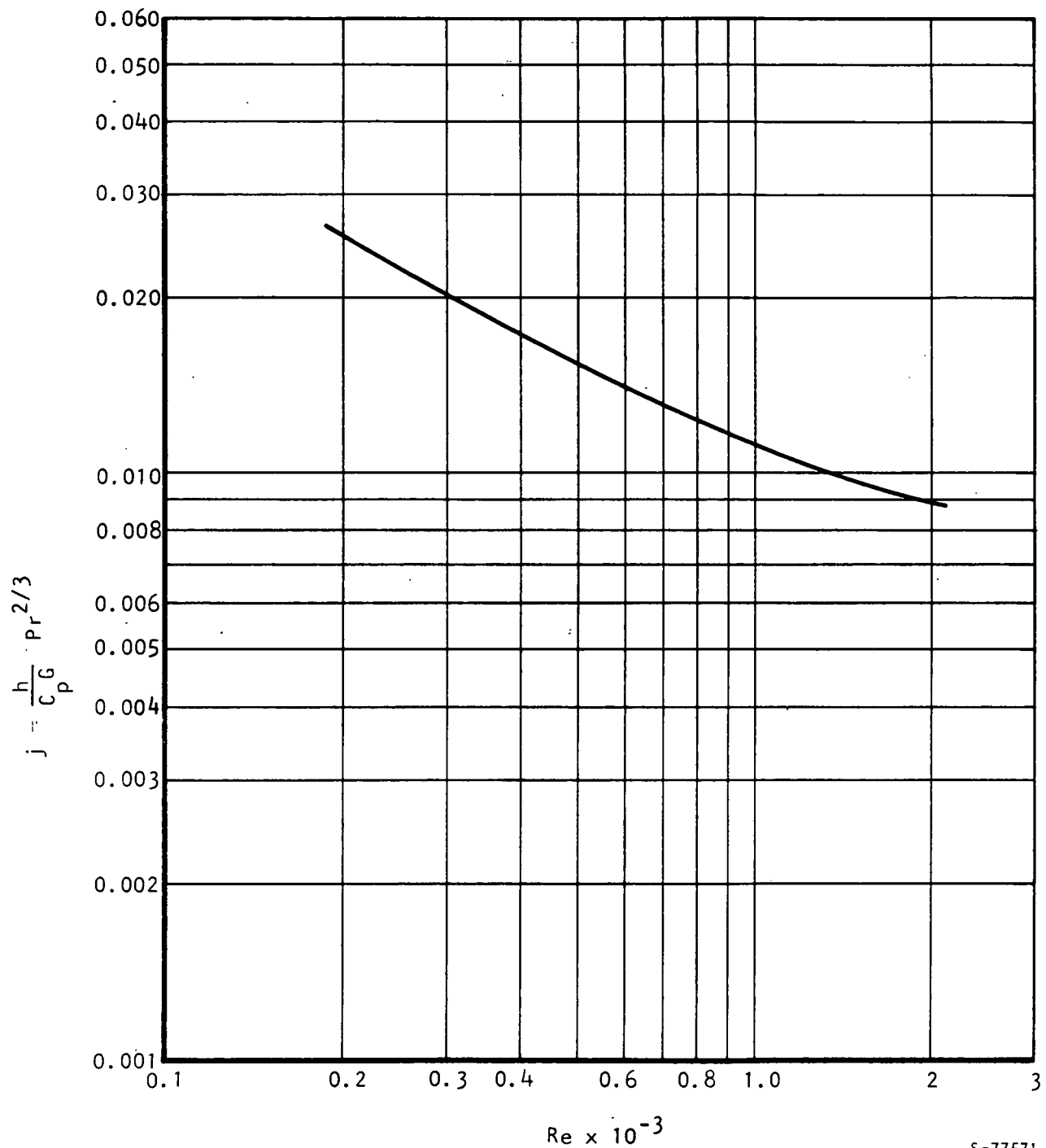
$\dot{m}$ = flow rate

Reynolds number is defined as

$$Re = \frac{D_H G}{\mu} \quad (3-11)$$

Figure 3-19 gives the Colburn j factor for the fin used in the GSFC VM refrigerator sump heat exchanger. This surface has 787 fins/m (20 fins/in.), an 0.00254 m (0.1 in.) offset length, fin length of 0.000762 m (0.030 in.) and a fin thickness of 0.0001016 m (0.004 in.). Figure 3-20 gives the friction factor for this surface. The pressure drop is then computed by use of

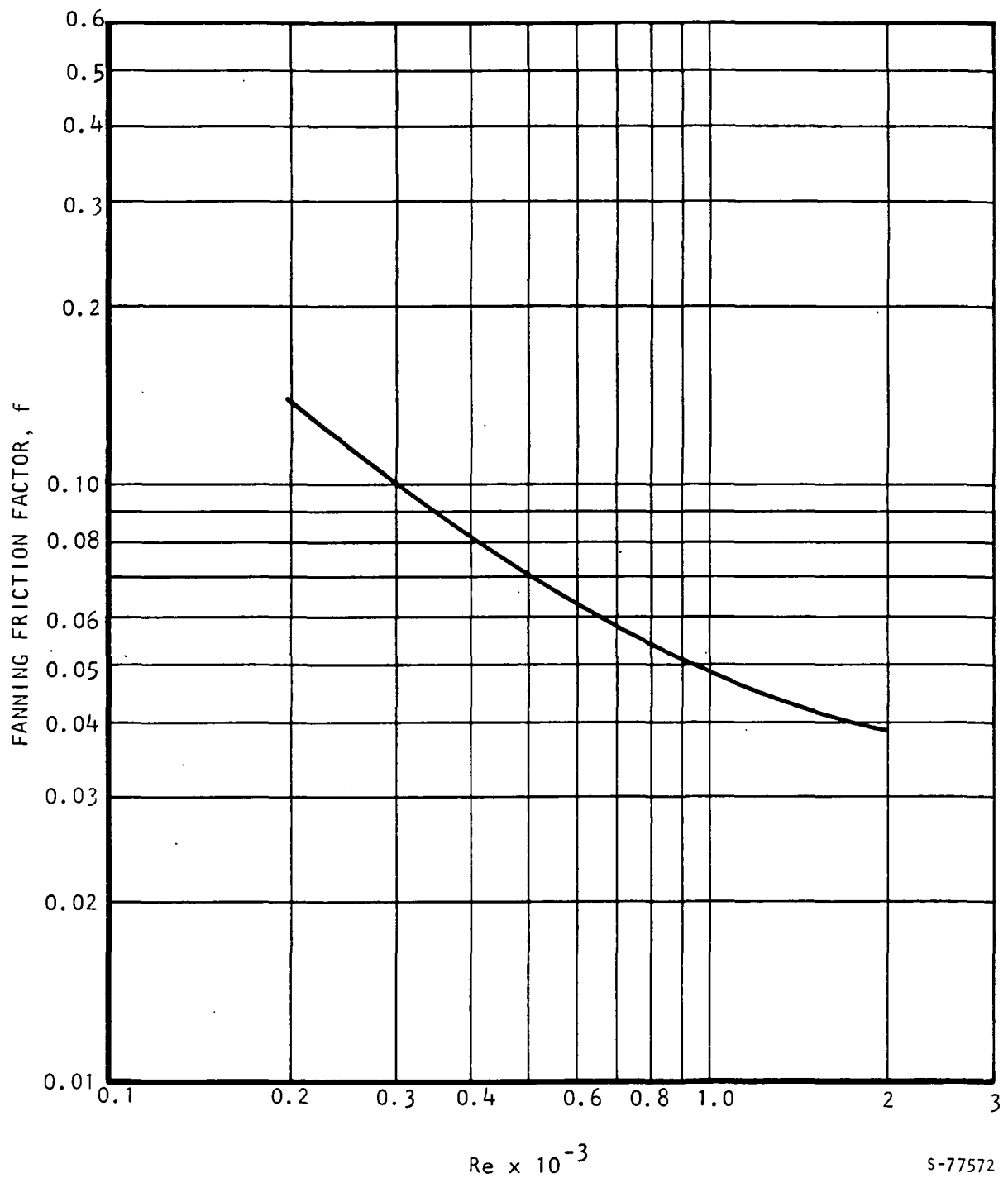




S-77571

Figure 3-19. Colburn j Factor vs Reynolds Number for Sump Heat Exchanger Heat Transfer Surface





S-77572

Figure 3-20. Fanning Friction Factor vs Reynolds Number for Sump Heat Exchanger Heat Transfer Surface



$$\Delta P = \frac{4fL}{D_H} \left(\frac{V^2}{2g_c} \right) \rho \quad (3-12)$$

where V = gas velocity

ρ = gas density

g_c = gravitational constant

4. Performance Characteristics

The sump heat exchanger performance characteristics are summarized in Table 3-9. The heat transfer performance is based on the average flow in each section of the heat exchanger. For the pressure drop, maximum flows were assumed. The lengths of the two sections of the heat exchanger have been chosen to produce a heat transfer conductance ratio in proportion to the ratio of flows in the two sections. The film temperature drop at a heat load of 80 watts is 2.96°K (5.33°R).

TABLE 3-9

SUMP HEAT EXCHANGER DESIGN AND PERFORMANCE SUMMARY

Parameter	Section 1	Section 2
Total Area, m^2 (ft^2)	0.01475 (0.1588)	0.00794 (0.0855)
Maximum Flow, kg/sec (lb/sec)	0.00329 (0.00724)	0.001237 (0.00272)
Average Flow, kg/sec (lb/sec)	0.00208 (0.00458)	0.000764 (0.00168)
Fin Effectiveness	0.837	0.884
Fluid Temperature, $^{\circ}\text{K}$ ($^{\circ}\text{R}$)	344 (620)	344 (620)
Heat Transfer Coefficient, $\text{watts}/\text{m}^2\text{-}^{\circ}\text{K}$ ($\text{Btu}/\text{ft}^2\text{-}^{\circ}\text{R hr}$)	1489 (262)	991 (174.2)
Conductance (ηhA), $\text{watts}/^{\circ}\text{K}$ ($\text{watts}/^{\circ}\text{R}$)	19.71 (10.95)	7.30 (4.06)
Pressure Drop, N/m^2 (psi)	429 (0.0623)	52.7 (0.00765)
Conductance (hA), $\text{watts}/^{\circ}\text{K}$ ($\text{watts}/^{\circ}\text{R}$)	Total for Sections 1 and 2	
	27.01 (15.01)	



Cold End Insulation

1. Introduction

The cold end insulation functions to limit the heat transferred from the ambient atmosphere to the cryogenic cold end of the VM refrigerator. The insulation system analysis is somewhat unique as compared to the more common problem of insulating a constant temperature cryogenic heat sink from the constant temperature ambient heat source. The cold finger of the VM is composed of two regions: (1) a constant temperature portion comprised of the cold end heat exchanger and the refrigeration load mounting surface and (2) the cold regenerator outer wall, which has a linear temperature gradient imposed on it. This linear gradient varies from the cold end temperature to the sump temperature, which is above ambient temperature.

In discussions between NASA/GSFC, AiResearch, and Honeywell Radiation Center personnel, further characteristics of the cold end insulation scheme were agreed upon. Since the VM refrigerator is a flight prototype machine, it is desirable to be able to evacuate the cold end vacuum enclosure and seal it off without the requirement for further pumping at a later date. This requirement dictates not only a leak tight enclosure but also the use of materials which do not outgas with time under hard vacuum. Thus the use of the cryogenic superinsulations (aluminized mylar) is precluded, along with any other organic shields supported by fiberglass pads.

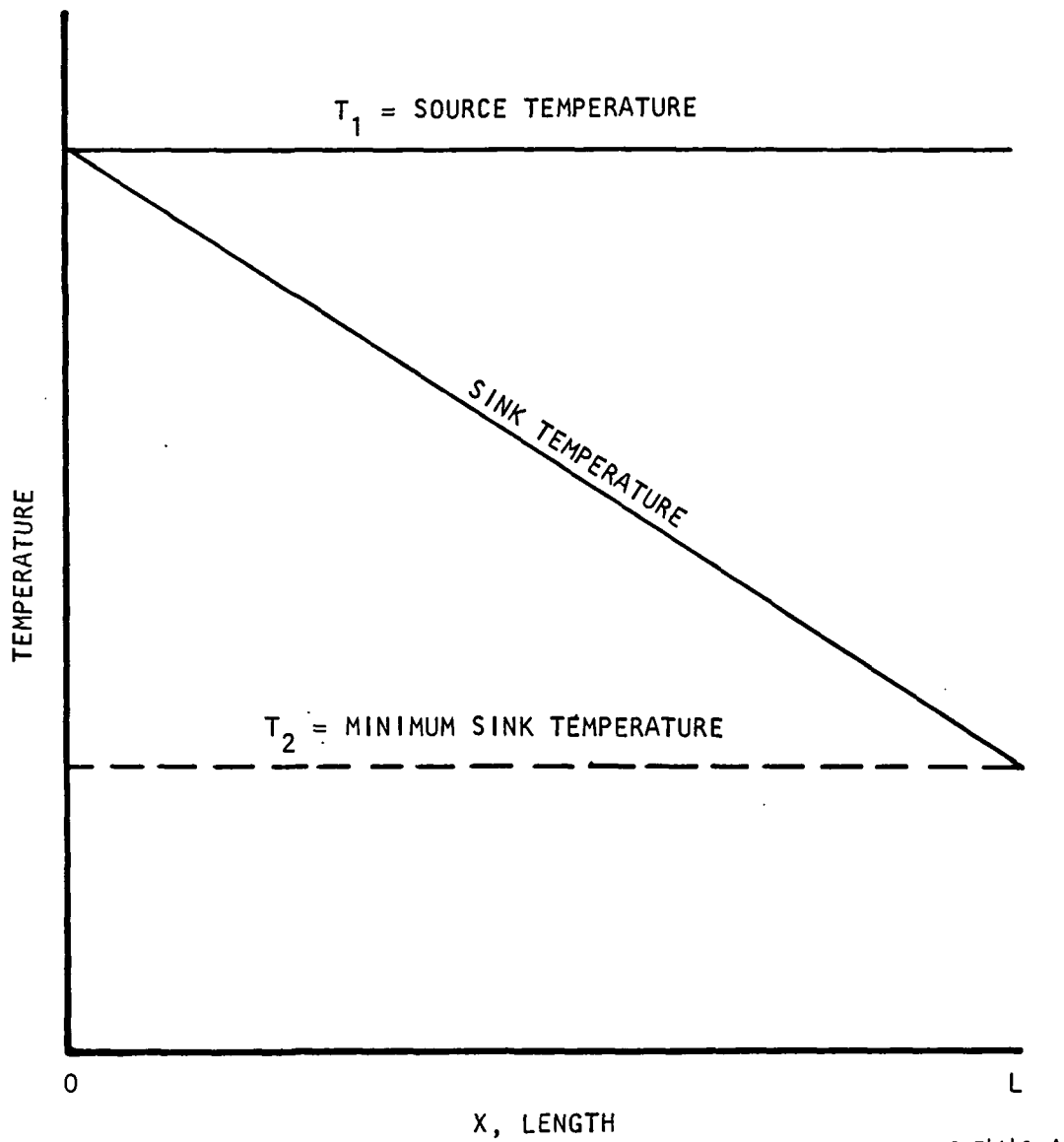
2. Preliminary Analysis

The analysis is divided into two parts: determination of the radiation heat transfer from a constant temperature source (the heat shield) to a sink with a linear temperature gradient (the regenerator body and cold end heat exchanger) and determination of the heat shield temperature. The output of the regenerator analysis computer program (Table 3-7) indicates a linear temperature gradient along the sink. In addition, axial conduction along the metallic radiation shield was expected to result in a constant temperature at the shield. The outer wall of the vacuum enclosure is at a constant temperature of 294°K (530°R). In view of the regenerator temperature profile and considerations discussed above, the cold-end insulation analysis follows.

The geometric view factor between two concentric cylinders is developed in a number of heat transfer textbooks. However, radiation between two surfaces, where a temperature gradient is imposed on one, is generally not discussed. Figure 3-21 shows the model used to determine the proper heat sink temperature for radiation calculations for this case. The radiation heat transfer is expressed as:

$$dQ = \frac{\sigma F_{1-2} A_1}{L} (T_1^4 - T_{\text{sink}}^4) dx \quad (3-13)$$





S-74142-A

$$T_X = T_{\text{SINK}} = T_1 - MX; \text{ WHERE } M = \text{SLOPE}$$

Figure 3-21. Cold End Temperature Profile for Heat Leak Calculations



where Q = heat transfer rate

$F_{1-2}A_1$ = emissivity view factor area product

L = length of cold cylinder

T_{sink} = local sink temperature $f(x)$

substitution of the expression for sink temperature results in

$$dQ = \frac{\sigma F_{1-2}A_1}{L} \left[T_1^4 - (T_1 - mx)^4 \right] dx \quad (3-14)$$

Expanding and integrating between the limits of $x = 0$ and $x = L$ results in the following expression for the total heat transfer by radiation

$$Q = \sigma F_{1-2}A_1 \left[2T_1^3 mL - 2T_1^2 m^2 L^2 + T_1 m^3 L^3 - \frac{m^4 L^4}{5} \right] \quad (3-15)$$

Noting that $mL = T_1 - T_2$, the following expression may be obtained.

$$Q = F_{1-2}A_1 \left[.8T_1^4 - .2T_1^3T_2 - .2T_1^2T_2^2 - .2T_1T_2^3 + T_2^4 \right] \quad (3-16)$$

The radiation heat transfer driving force is defined as $\left[(T_{\text{source}})^4 - (T_{\text{sink}})^4 \right]$. The expression for the proper average sink temperature with the conditions of Figure 3-21 may be obtained from Equation 3-16.

$$(T_{\text{sink average}})^4 = 0.2 \left[T_1^4 + T_1^3T_2 + T_1^2T_2^2 + T_1T_2^3 \right] - T_2^4 \quad (3-17)$$

Equation 3-16 permits determination of the radiation heat transfer from a constant temperature source to a heat sink with a linear temperature gradient. It is interesting to note that if the ambient temperature of 294°K (530°R) and the cold end temperature of 62.3°K (112°R) are substituted into Equation 3-16 (no radiation heat shield) the resulting heat transfer is approximately 75 percent of that which would occur if the entire regenerator wall were at the coldest temperature, 62.3°K (112°R).

Verification of the constant temperature of a single radiation shield was considered next. The shield is suspended in the vacuum annulus by fiberglass pads, and is thus not in contact with any metallic components of the VM. Ignoring the conduction through the fiberglass (a non-conservative assumption) the shield receives heat from the outer skin, and rejects it to the regenerator wall, by radiation alone. Near the cold end of the machine, the shield is cooled by exposure to the low temperature of the cold finger. Near the sump end, where the outer wall and regenerator wall are at the same temperature,



there is no temperature potential for heat transfer to or from the shield. Since the shield is a good heat conductor, the colder end will cool the warm end, until overall equilibrium is reached. This complex problem was not solved by hand, but by use of a thermal analyzer computer program.

A thirty-node network was utilized. The ten nodes along the outer wall were held at a constant temperature of 294°K (530°R). The ten nodes on the regenerator wall were held at the linear temperature gradient imposed by the gas flowing through the regenerator. The ten nodes on the radiation heat shield were linked to each other by a conduction resistance resulting from a shield thickness of 0.010 inches manufactured from aluminum. The shield nodes were also linked to the inner and outer walls by radiation resistances which were computed using emissivity of all surfaces as 0.05.

The solution indicated that the radiation heat shield reached a constant temperature of 268°K (482°R). The axial temperature variation was less than 0.566°K (1°R). The total cold end heat leak calculated for this configuration is 68 milliwatts. It should be recalled that the conduction through the fiberglass supports to the radiation shield was neglected. This additional heat input to the heat shield would increase the shield temperature and thus the overall heat leak. Thus the 68 mw is a theoretical value that is unattainable in actual practice.

The preliminary analysis also ignored the portion of the regenerator wall which is hotter than ambient. In view of the heat leak calculated above and the shortcomings of the thermal model, it was decided to utilize an insulation configuration employing two heat shields, and to employ a more detailed thermal model.

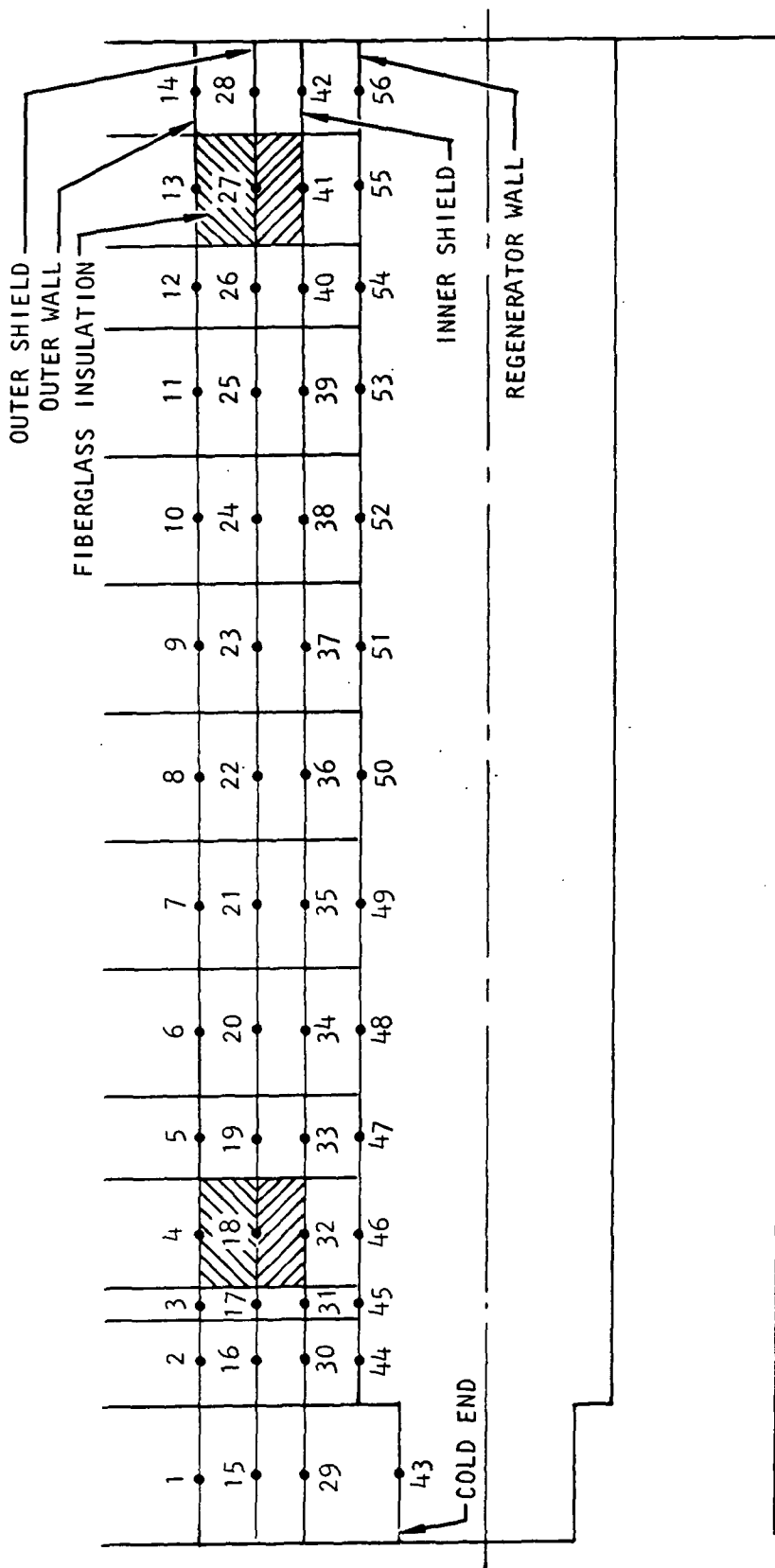
3. Detailed Analysis

In addition to the preliminary results described above, an analysis performed by NASA/GSFC indicated that two radiation heat shields would be required to adequately insulate the cold end of the GSFC fractional watt VM. Therefore a relatively detailed thermal model of the desired configuration was set up and solved by use of the AiResearch thermal analyzer computer program.

The 56-node thermal network utilized as a model is shown in Figure 3-22. The radiation heat shields were assumed to be made of aluminum with an emissivity of 0.05 and a shield thickness of 0.000254 m (0.010 in.). The emissivity of the outer wall, regenerator wall, and cold end was taken as 0.15. The areas of the various concentric cylinders were accounted for in calculating the emissivity view factors.

The thermal analyzer computer program was utilized to determine the steady state temperatures of the nodes. The program printout is presented in Table 3-10 and the resulting temperature distribution is plotted in Figure 3-23. The output data indicates that the radiation shields attain a constant temperature. Both shields attain equilibrium at a temperature below that of the outer wall and also below the temperature of the warm end of the regenerator wall. Thus the inner radiation shield receives heat from two sources; the heat leak





NODES 1 THRU 14: OUTER WALL, CONSTANT TEMPERATURE AT 294°K (530°R)

NODES 15 THRU 28: OUTER SHIELD. ALL NODES LINKED BY CONDUCTIVE RESISTANCE

NODES 29 THRU 42: INNER SHIELD. ALL NODES LINKED BY CONDUCTIVE RESISTANCE

NODES 43 AND 44: COLD END, CONSTANT TEMPERATURE AT 62.3°K (112°R)

NODES 45 THRU 56: REGENERATOR WALL, LINEAR TEMPERATURE GRADIENT FROM 62.3 (112) TO 344°K (620°R)

OUTER WALL AND BOTH SHIELDS LINKED BY CONDUCTIVE RESISTANCES THRU FIBERGLASS

OUTER WALL, BOTH SHIELDS, COLD END, AND REGENERATOR WALL LINKED BY RADIATIVE RESISTANCES EXCEPT AT FIBERGLASS

S-74358-A

Figure 3-22. Fractional Watt VM Cold Finger Thermal Analyzer Model

TABLE 3-10

THERMAL ANALYZER SOLUTION OF COLD END INSULATION
WITH ALUMINUM HEAT SHIELDS

STEADY STATE SOLUTION

NO.OF ITER.= 398

DTEMP= .0007

ACCEL= .0000

NODE NO.	TEMP.	HEAT IN.	RHOV	CPN	KN	TEMP.
1	530.00	.0000	-1.000000	.2000	100.00000	294.44
2	530.00	.0000	-1.000000	.2000	100.00000	294.44
3	530.00	.0000	-1.000000	.2000	100.00000	294.44
4	530.00	.0000	-1.000000	.2000	100.00000	294.44
5	530.00	.0000	-1.000000	.2000	100.00000	294.44
6	530.00	.0000	-1.000000	.2000	100.00000	294.44
7	530.00	.0000	-1.000000	.2000	100.00000	294.44
8	530.00	.0000	-1.000000	.2000	100.00000	294.44
9	530.00	.0000	-1.000000	.2000	100.00000	294.44
10	530.00	.0000	-1.000000	.2000	100.00000	294.44
11	530.00	.0000	-1.000000	.2000	100.00000	294.44
12	530.00	.0000	-1.000000	.2000	100.00000	294.44
13	530.00	.0000	-1.000000	.2000	100.00000	294.44
14	530.00	.0000	-1.000000	.2000	100.00000	294.44
15	509.55	.0000	.010000	.2000	100.00000	283.08
16	509.55	.0000	.010000	.2000	100.00000	283.08
17	509.55	.0000	.010000	.2000	100.00000	283.08
18	509.55	.0000	.010000	.2000	100.00000	283.08
19	509.55	.0000	.010000	.2000	100.00000	283.08
20	509.56	.0000	.010000	.2000	100.00000	283.09
21	509.56	.0000	.010000	.2000	100.00000	283.09
22	509.56	.0000	.010000	.2000	100.00000	283.09
23	509.56	.0000	.010000	.2000	100.00000	283.09
24	509.55	.0000	.010000	.2000	100.00000	283.09
25	509.55	.0000	.010000	.2000	100.00000	283.08
26	509.54	.0000	.010000	.2000	100.00000	283.08
27	509.54	.0000	.010000	.2000	100.00000	283.08
28	509.54	.0000	.010000	.2000	100.00000	283.08
29	471.06	.0000	.010000	.2000	100.00000	261.70
30	471.09	.0000	.010000	.2000	100.00000	261.72
31	471.12	.0000	.010000	.2000	100.00000	261.73
32	471.17	.0000	.010000	.2000	100.00000	261.76
33	471.23	.0000	.010000	.2000	100.00000	261.80
34	471.32	.0000	.010000	.2000	100.00000	261.84
35	471.47	.0000	.010000	.2000	100.00000	261.93
36	471.66	.0000	.010000	.2000	100.00000	262.03
37	471.87	.0000	.010000	.2000	100.00000	262.15
38	472.11	.0000	.010000	.2000	100.00000	262.20
39	472.35	.0000	.010000	.2000	100.00000	262.42
40	472.51	.0000	.010000	.2000	100.00000	262.51
41	472.65	.0000	.010000	.2000	100.00000	262.58
42	472.72	.0000	.010000	.2000	100.00000	262.62
43	112.00	.0000	-1.000000	.2000	100.00000	62.22
44	112.00	.0000	-1.000000	.2000	100.00000	62.22
45	118.40	.0000	-1.000000	.2000	100.00000	65.78
46	145.00	.0000	-1.000000	.2000	100.00000	80.56
47	183.00	.0000	-1.000000	.2000	100.00000	101.67
48	226.00	.0000	-1.000000	.2000	100.00000	125.56
49	277.00	.0000	-1.000000	.2000	100.00000	153.89
50	328.00	.0000	-1.000000	.2000	100.00000	182.22
51	378.00	.0000	-1.000000	.2000	100.00000	210.00
52	430.00	.0000	-1.000000	.2000	100.00000	238.89
53	480.00	.0000	-1.000000	.2000	100.00000	266.67
54	523.00	.0000	-1.000000	.2000	100.00000	290.56
55	562.00	.0000	-1.000000	.2000	100.00000	312.22
56	601.00	.0000	-1.000000	.2000	100.00000	333.69

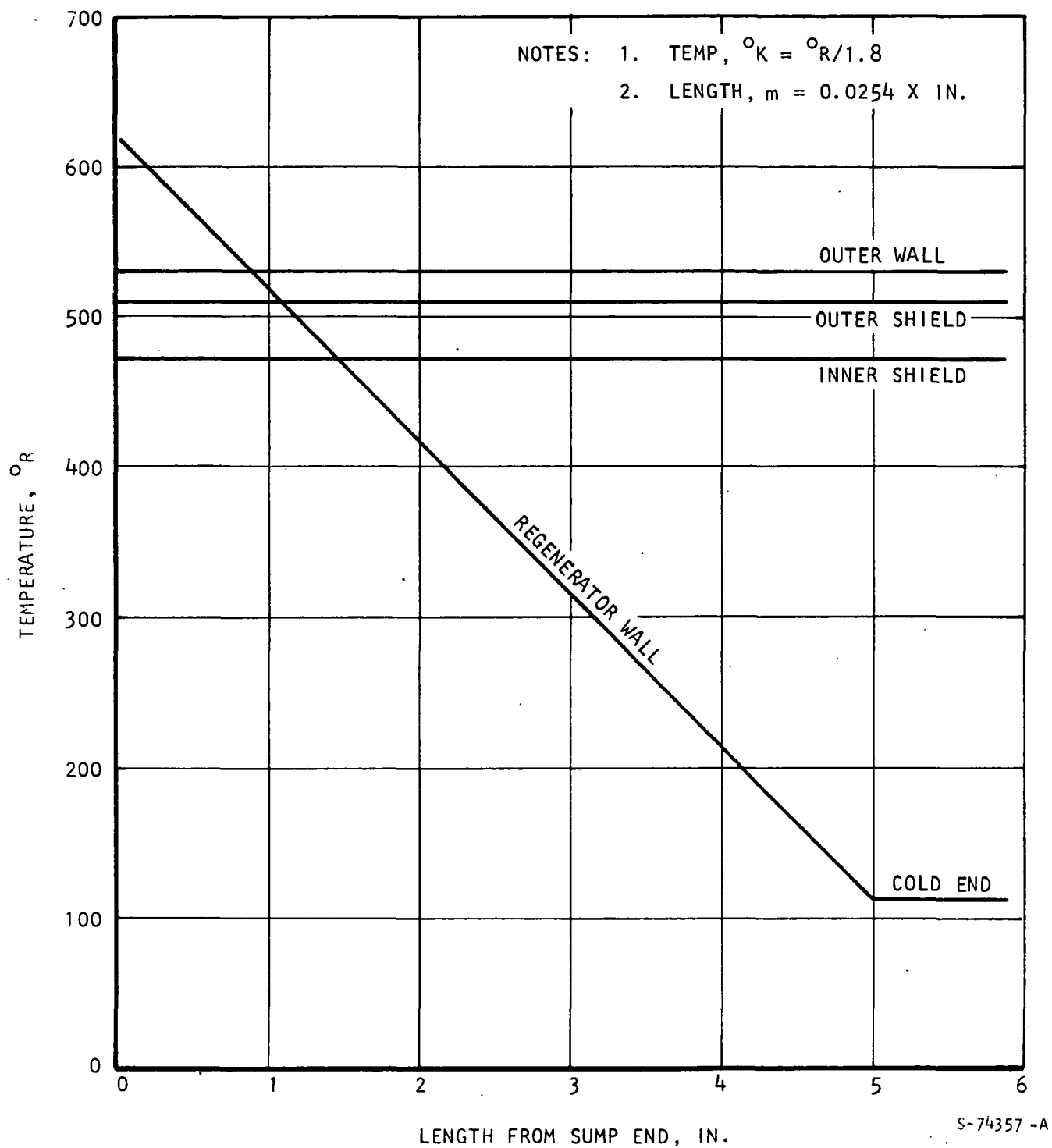


Figure 3-23. Temperature Distribution, Cold Finger of Fractional Watt VM



from ambient and from the warm end of the cold regenerator. The heat from these two sources is radiated to the cold end of the machine, and to the colder portion of the regenerator wall.

The heat leak to the cold portion of the regenerator was determined by two methods. The first method was summing the heat flow at the individual nodes; the second involved the use of the average sink temperature expression (Equation 3-17). The two methods yield identical results, thus indicating that the number of nodes utilized in the thermal network is sufficient.

The calculated heat leak is summarized in Table 3-11.

TABLE 3-11
COLD END INSULATION HEAT LEAK WITH
ALUMINUM HEAT SHIELDS

Portion of Cold Finger	Heat Leak, milliwatts
Cold End	21.9
Regenerator from Cold End to Point Where $T_{\text{wall}} = T_{\text{shield}}$	<u>73.3</u>
Total	95.2

Of the total heat leak, shown in Table 3-11, 60.7 milliwatts is contributed by the ambient environment (outer wall) and 34.5 milliwatts is re-radiated from that portion of the regenerator wall which is hotter than the inner heat shield. The re-radiation portion of the total heat leak will exist no matter how many shields are used.

The heat shields may be modified to decrease the re-radiation slightly, but no significant reduction is achieved unless a very complex network of radial baffles and shields are utilized. One possible method of decreasing the heat leak investigated was reduction of the shield thermal conductivity. This has the effect of reducing the inner shield temperature at the cold end of the machine, and thus the radiation to the cold end. However, the temperature of the shield at the warm end is also increased. This has the effect of increasing both the temperature potential to the regenerator wall and exposing more of the regenerator to the higher temperature.

In order to evaluate this effect, the shield thermal conductivity was changed from that of aluminum to that of stainless steel. This type of shield construction would involve an aluminum surface deposited on a low thermal conductivity core. The total heat leak with this configuration is 93.5 milliwatts. The added complexity of construction is not warranted for this decrease in heat leak.



The heat shields could also be shortened at the hot end of the regenerator, in order to decrease the re-radiation. This concept was not investigated in detail. The shortened heat shields would have to be isolated from the hot portion of the regenerator by radial baffles in order to prevent their "seeing" the high temperature. The radial baffles would, in turn, have to be thermally isolated from the heat shields in order to eliminate conduction. The resulting network of baffles and shields would be extremely complex, and very difficult to manufacture.

Another attractive method of decreasing the cold end heat leak is reduction of the heat shield emissivity. Aluminum was utilized in this analysis, because of its availability, ease of manufacturing operations, and low emissivity. Other materials (gold, silver, and rhodium, for example) exhibit emissivity values considerably lower than that of aluminum. The primary problem with silver and gold is oxidation of the highly polished surfaces. Rhodium is especially attractive from the standpoint of corrosion resistance. It has been used as the reflector material for carbon-arc searchlights, and may be readily cleaned by hand without damage to the bright surface. Thus the special handling required of highly polished gold and silver surfaces is not required. In addition, rhodium is easily plated onto base materials.

The room temperature emissivity of rhodium is quoted as 0.02 (General Electric Heat Transfer data handbook, Section G515.5, pg. 24). The thermal analyzer computer program was utilized to solve the thermal network of Figure 3-22, with the emissivities of both heat shields and the outer vacuum enclosure equal to 0.02. The resulting temperature distribution is presented in Table 3-12. The temperatures do not differ greatly from those of the aluminum heat shield configuration. However, the heat transmitted is considerably lower due to the lower emissivity. The calculated heat leak for this configuration is summarized in Table 3-13.

This heat leak represents a more reasonable value than that obtained with aluminum heat shields. Thus the rhodium plating will yield an acceptable cold end insulation scheme.

Hot End Insulation

1. Introduction

The hot end insulation limits the heat transferred from the heater, the hot end pressure dome, and the hot regenerator outer wall to the ambient atmosphere surrounding the VM refrigerator. It is essential to limit this hot end loss to a minimum practical value, as heat loss in this manner is a direct loss of input power.

2. Analysis

Originally, a multiple radiation shield insulation scheme was visualized for the hot end. Accordingly, an analysis of the hot regenerator wall was performed similar to that for the cold end. The temperature gradient on the hot regenerator wall is linear, but the relative roles of the heat source and heat sink are reversed from the cold end. The heat sink (the outer vacuum enclosure)



TABLE 3-12

THERMAL ANALYZER SOLUTION OF COLD-END INSULATION NETWORK
WITH RHODIUM-PLATED HEAT SHIELDS

STEADY STATE SOLUTION						
NO. OF ITER. = 794 DTMP = .0008 ACCEL = .0000						
NODE NO.	TEMP.	HEAT IN.	RHOV	CPN	KT	TEMP.
1	530.00	.0000	-1.000000	.2000	100.00000	294.44
2	530.00	.0000	-1.000000	.2000	100.00000	294.44
3	530.00	.0000	-1.000000	.2000	100.00000	294.44
4	530.00	.0000	-1.000000	.2000	100.00000	294.44
5	530.00	.0000	-1.000000	.2000	100.00000	294.44
6	530.00	.0000	-1.000000	.2000	100.00000	294.44
7	530.00	.0000	-1.000000	.2000	100.00000	294.44
8	530.00	.0000	-1.000000	.2000	100.00000	294.44
9	530.00	.0000	-1.000000	.2000	100.00000	294.44
10	530.00	.0000	-1.000000	.2000	100.00000	294.44
11	530.00	.0000	-1.000000	.2000	100.00000	294.44
12	530.00	.0000	-1.000000	.2000	100.00000	294.44
13	530.00	.0000	-1.000000	.2000	100.00000	294.44
14	530.00	.0000	-1.000000	.2000	100.00000	294.44
15	505.86	.0000	.010000	.2000	100.00000	281.03
16	505.86	.0000	.010000	.2000	100.00000	281.03
17	505.86	.0000	.010000	.2000	100.00000	281.03
18	505.86	.0000	.010000	.2000	100.00000	281.03
19	505.86	.0000	.010000	.2000	100.00000	281.03
20	505.86	.0000	.010000	.2000	100.00000	281.03
21	505.86	.0000	.010000	.2000	100.00000	281.03
22	505.86	.0000	.010000	.2000	100.00000	281.03
23	505.86	.0000	.010000	.2000	100.00000	281.03
24	505.86	.0000	.010000	.2000	100.00000	281.03
25	505.86	.0000	.010000	.2000	100.00000	281.03
26	505.86	.0000	.010000	.2000	100.00000	281.03
27	505.86	.0000	.010000	.2000	100.00000	281.03
28	505.86	.0000	.010000	.2000	100.00000	281.03
29	472.45	.0000	.010000	.2000	100.00000	262.47
30	472.47	.0000	.010000	.2000	100.00000	262.48
31	472.49	.0000	.010000	.2000	100.00000	262.49
32	472.51	.0000	.010000	.2000	100.00000	262.51
33	472.54	.0000	.010000	.2000	100.00000	262.52
34	472.56	.0000	.010000	.2000	100.00000	262.54
35	472.65	.0000	.010000	.2000	100.00000	262.58
36	472.74	.0000	.010000	.2000	100.00000	262.63
37	472.85	.0000	.010000	.2000	100.00000	262.69
38	472.97	.0000	.010000	.2000	100.00000	262.76
39	473.10	.0000	.010000	.2000	100.00000	262.83
40	473.18	.0000	.010000	.2000	100.00000	262.88
41	473.25	.0000	.010000	.2000	100.00000	262.92
42	473.30	.0000	.010000	.2000	100.00000	262.94
43	112.00	.0000	-1.000000	.2000	100.00000	62.22
44	112.00	.0000	-1.000000	.2000	100.00000	62.22
45	118.40	.0000	-1.000000	.2000	100.00000	65.73
46	145.00	.0000	-1.000000	.2000	100.00000	80.56
47	163.00	.0000	-1.000000	.2000	100.00000	101.67
48	226.00	.0000	-1.000000	.2000	100.00000	125.56
49	277.00	.0000	-1.000000	.2000	100.00000	153.89
50	328.00	.0000	-1.000000	.2000	100.00000	182.22
51	378.00	.0000	-1.000000	.2000	100.00000	210.00
52	430.00	.0000	-1.000000	.2000	100.00000	238.89
53	480.00	.0000	-1.000000	.2000	100.00000	266.67
54	523.00	.0000	-1.000000	.2000	100.00000	290.56
55	562.00	.0000	-1.000000	.2000	100.00000	312.22
56	601.00	.0000	-1.000000	.2000	100.00000	333.89

is at a constant temperature, while the temperature of the heat source (the regenerator outer wall) varies linearly. The temperature model utilized in the analysis is shown in Figure 3-24.

TABLE 3-13
HEAT LEAK OF FRACTIONAL WATT VM COLD FINGER WITH
RHODIUM PLATED HEAT SHIELDS

Heat Source	Heat Leak, milliwatts
Reradiation	17.0
Radiation from Outer Wall	15.0
Conduction from Outer Wall	<u>15.0</u>
Total	47.0

The analysis followed that performed for the cold end, with the substitution of the source temperature into Equation 3-13. The results indicate that the proper source temperature for use in the radiation calculations is represented by Equation 3-18.

$$\left(T_{\text{source average}}\right)^4 = 0.2 \left(T_1^4 + T_1^3 T_2 + T_1^2 T_2^2 + T_1 T_2^3 + T_2^4\right) \quad (3-18)$$

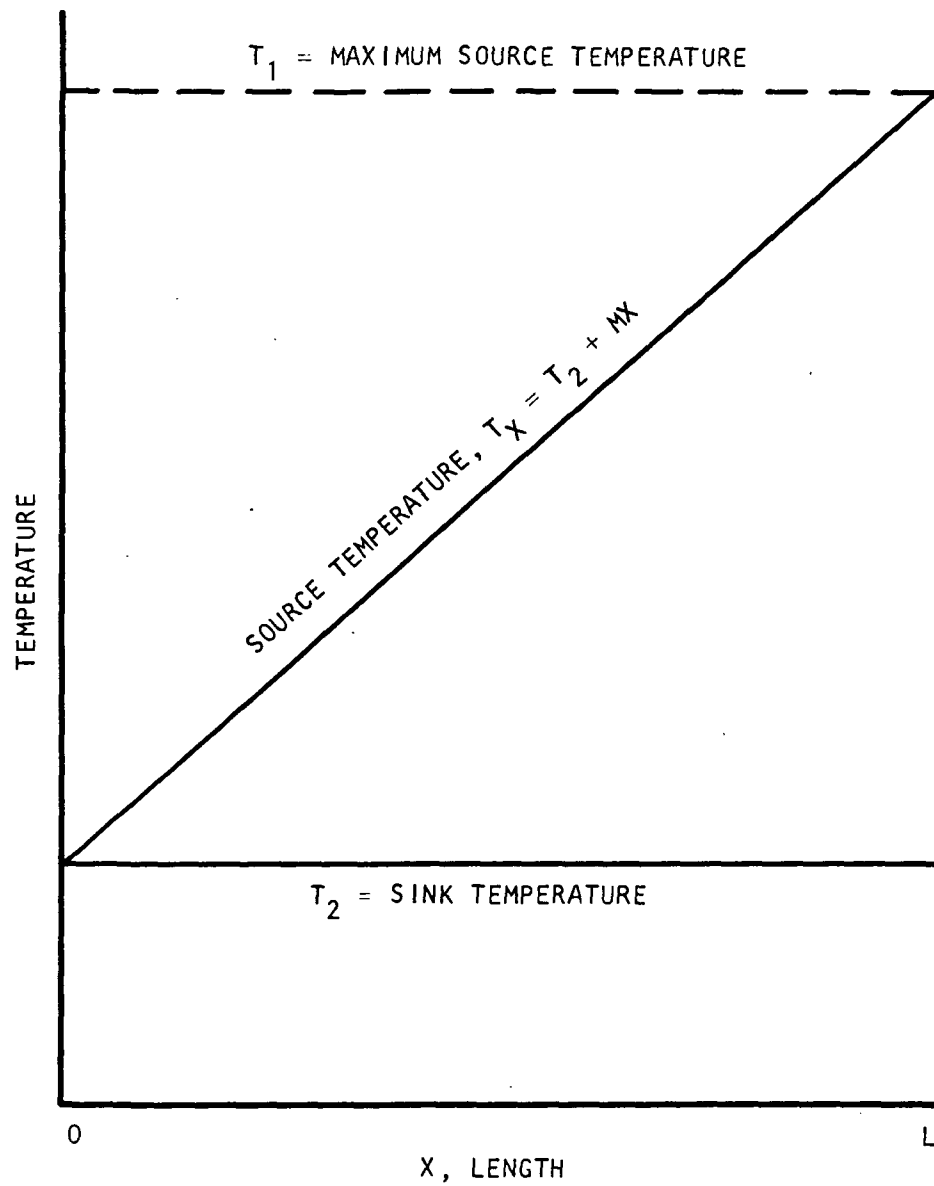
where T_1 = maximum source temperature

T_2 = sink temperature

This analysis indicates that the heat leak is approximately only 30 percent of that which would occur if the entire regenerator wall were at the maximum temperature.

While this result appears attractive, the constant temperature of the heat shield, as evidenced by the cold end insulation analysis, is not. The inner heat shield would attain a temperature near that of the constant temperature of the hot end dome and heater. This would then result in reradiation from the hot shield to the cooler regenerator wall, thus introducing an additional heat leak. The re-radiation would manifest itself as an additional, undesirable internal heat leak from the hot end to the sump region. Thus, while external heat losses would be low, internal losses would occur. This difficulty could be overcome by thermally grounding the heat shields at the sump end, thus imposing a linear temperature gradient on the shields similar to that on the regenerator wall. In addition, radial baffles would be required at frequent intervals on the inner shield to prevent the regenerator wall from seeing





S-77533

Figure 3-24. Hot End Temperature Profile for Radiation Heat Leak Calculations



higher temperature portions of the shield. Thermal grounding of the shields would provide additional heat leak paths from the hot end, also increasing the total heat leak and thus the thermal power input required.

An acceptable insulation configuration utilizing heat shields could be designed, but would be very complex and difficult to manufacture. Therefore, a composite insulation system utilizing fiberglass and Min-K has been utilized, similar to that incorporated into the GSFC 5 watt VM. This type of insulation system is attractive for the fractional watt VM, since the heater is bonded directly to the hot end dome. The heater thus operates at essentially the hot end temperature, as compared to that on the 5 watt machine which transfers its energy by radiation. The insulation system has been designed so that the temperature at the interface between the Min-K and the fiberglass is 742°K (1335°R).

This system is attractive from an assembly standpoint, and the calculated heat leak to ambient is 5.9 watts. This is compatible with the overall power input requirements for the fractional watt VM. The insulation system is shown on the preliminary design outline drawing.

Pressure Drop and Void Volume Tradeoff Factors

1. Introduction

Many components of the VM refrigerator require design tradeoffs between the introduction of either void volume or working fluid pressure drop into the system. Void volume is defined as any volume within the machine which does not vary with crankshaft rotation. Since both of these factors detract from the refrigeration capacity of the machine, relative weighting factors for their effects are desirable. Unfortunately, minimization of one factor leads to large values of the other. Thus, if the pressure drop of a particular flow passage is made very low, the void volume becomes large. Conversely, if the void volume is minimized, pressure drop becomes relatively large. In addition, the effect of void volume depends on the temperature zone of the machine in which this volume occurs.

The first step in the evaluation of the tradeoff factors was modification of the ideal cycle computer program.

2. Modifications to the Ideal Computer Program

The VM refrigerator ideal cycle computer program, K0270, was modified to simplify its use. The program now has the capability to vary any of the seven volumes that are input over any desired range, with any desired increment. The charge pressure or maximum cycle pressure are now alternate inputs, and whichever one of these is chosen may be varied in the same manner as the volumes. The maximum hot and cold end pressure drops are now also inputs. The new input and description is presented in Table 3-14.

For each parameter that may be varied, there are three inputs; the initial value, the increment, and the final value. The final value does not have to



TABLE 3-14
INPUT FOR K0270

```

10 READ (5,3) HEAD
   READ(5,1) TCO,TC,TA,TH,TRC,TRH,R,DPH
   READ(5,1) ZCO,ZC,ZA,ZH,ZRC,ZRH,SPEED,DPC
   READ(5,1) VCD,VCDI,VCDL,VAD,VADI,VADL
   READ(5,1) VHD,VHDI,VHDL,VRC,VRCI,VRCL
   READ(5,1) VRH,VRHI,VRHL,VCO,VCOI,VCOL
   READ(5,1) VHO,VHOI,VHOL,PCO,PCOI,PCOL
   READ(5,2) NPRINT, IDEBUG
      INPUT PARAMETERS
      TCO    =CHARGE GAS TEMP.(R)
      TC     =COLD VOL. TEMP.(R)
      TA     =SUMP VOL. TEMP.(R)
      TH     =HOT VOL. TEMP.(R)
      TRC    =COLD REGEN. TEMP.(R)
      TRH    =HOT REGEN. TEMP.(R)
      R       =GAS CONSTANT (FT-LB/LBM-R)
      DPH    =HOT MAXIMUM PRESSURE DROP,PSI
      ZXX     =COMPRESSIBILITY AT TXX
      SPEED  =ENGINE SPEED (RPM)
      DPC    =COLD MAXIMUM PRESSURE DROP,PSI
      VCD     =COLD DEAD VOL. (CU-IN)
      VAD     =SUMP DEAD VOL. (CU-IN)
      VHD     =HOT DEAD VOL. (CU-IN)
      VRC     =COLD REGEN. VOL. (CU-IN)
      VRH     =HOT REGEN. VOL. (CU-IN)
      VCO     =COLD DISPLACED VOL. (CU-IN)
      VHO     =HOT DISPLACED VOL. (CU-IN)
      PCO     =MAXIMUM CYCLE PRESSURE,PSIA(IF MINUS,CHARGE PRESS)
      ---I    =INCREMENT
      ---L    =LAST VALUE
      NPRINT=NO. OF DEG. INCREMENT PRINT OUT IS DESIRED
      IDEBUG =ADDITIONAL PRINT OUT CONTROL
              =0  NO EXTRA PRINT OUT
              =1  EXTRA PRINT OUT

```



be an even number of increments from the initial value. The program only checks to see that the new current value is not greater than the final value. If a zero or negative increment is input, the program will change it to a large positive value in order to prevent a large number of insignificantly small increments being calculated. Particular attention must be paid to this when input is charge pressure (negative values are input). As an example of this, it is desired to vary the charge pressure from 4.83×10^6 (700) to 5.51×10^6 N/m² (800 psia) in 6.895×10^4 N/m² (10 psi) increments. The initial value input would be -800, the increment is 10, and the final value is -700.

An attempt has been made to prevent inadvertently receiving very large amounts of output. Therefore, all possible combinations of all variables are not calculated. Thus if each of the eight variables is allowed to assume ten values, eighty cases will be calculated and not 10^8 . Each variable is incremented in the order of input until the final value has been exceeded. The variable is left at its final value and incrementing of the next variable is begun. It is therefore recommended that only one variable be incremented per run.

If the pressure input is negative (charge pressure) the program takes its absolute value and calculates the mass of gas in the system from charge pressure and temperature. If the input is positive (maximum cycle pressure) the program determines the crank angle at which maximum pressure occurs. This angle is determined by differentiating the expression for pressure as a function of crank angle and setting the derivative equal to zero. The program then determines the mass of gas in the system from maximum cycle pressure and the expression for pressure as a function of crank angle. Once the mass of gas has been determined, the cycle calculations are made in the same manner. The program output has been modified to include the angle at which maximum cycle pressure occurs.

If the pressure drops are input as zero or blank, the calculations are not changed from the ideal case. If a positive pressure drop is input, the cycle pressures are not modified in the output. The ideal refrigeration and heat input are changed. In order to accomplish this, the cycle pressures are modified before the $\int P dV$ is evaluated. A pressure drop is evaluated at each crank angle increment by multiplying the maximum pressure drop by the square of the ratio of flow rate and maximum flow rate. If flow is into the cold volume, the pressure drop is subtracted from the ideal pressure. If flow is out of the cold volume, the pressure drop is added to the ideal pressure. The addition or subtraction of pressure drop as a function of flow direction is reversed for the hot end. If this modification of the integrals has occurred, a message is printed.

With these modifications to the ideal cycle program completed, a study was performed to determine the effects of void volume and pressure drop on VM refrigerator performance.



"Page missing from available version"

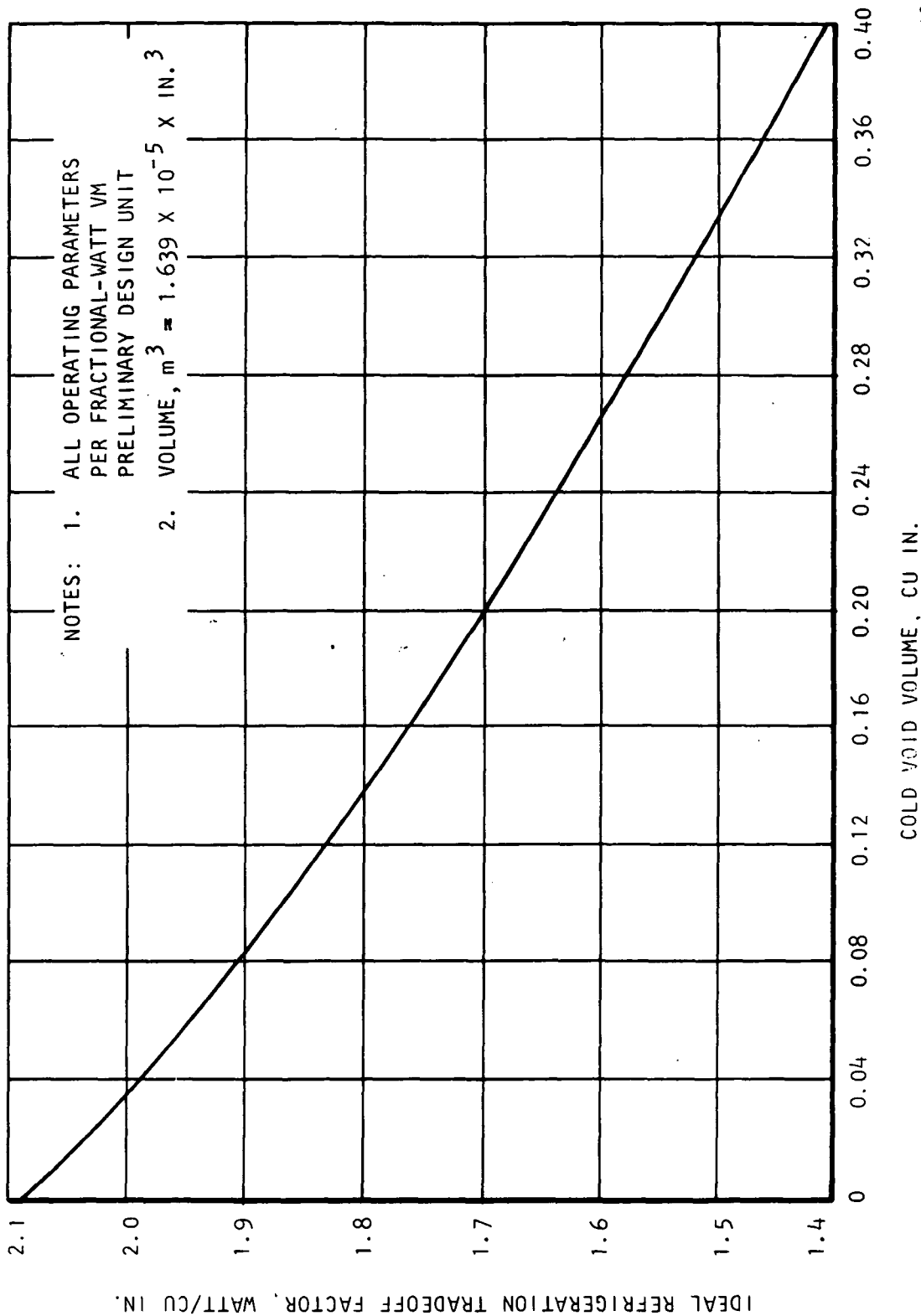
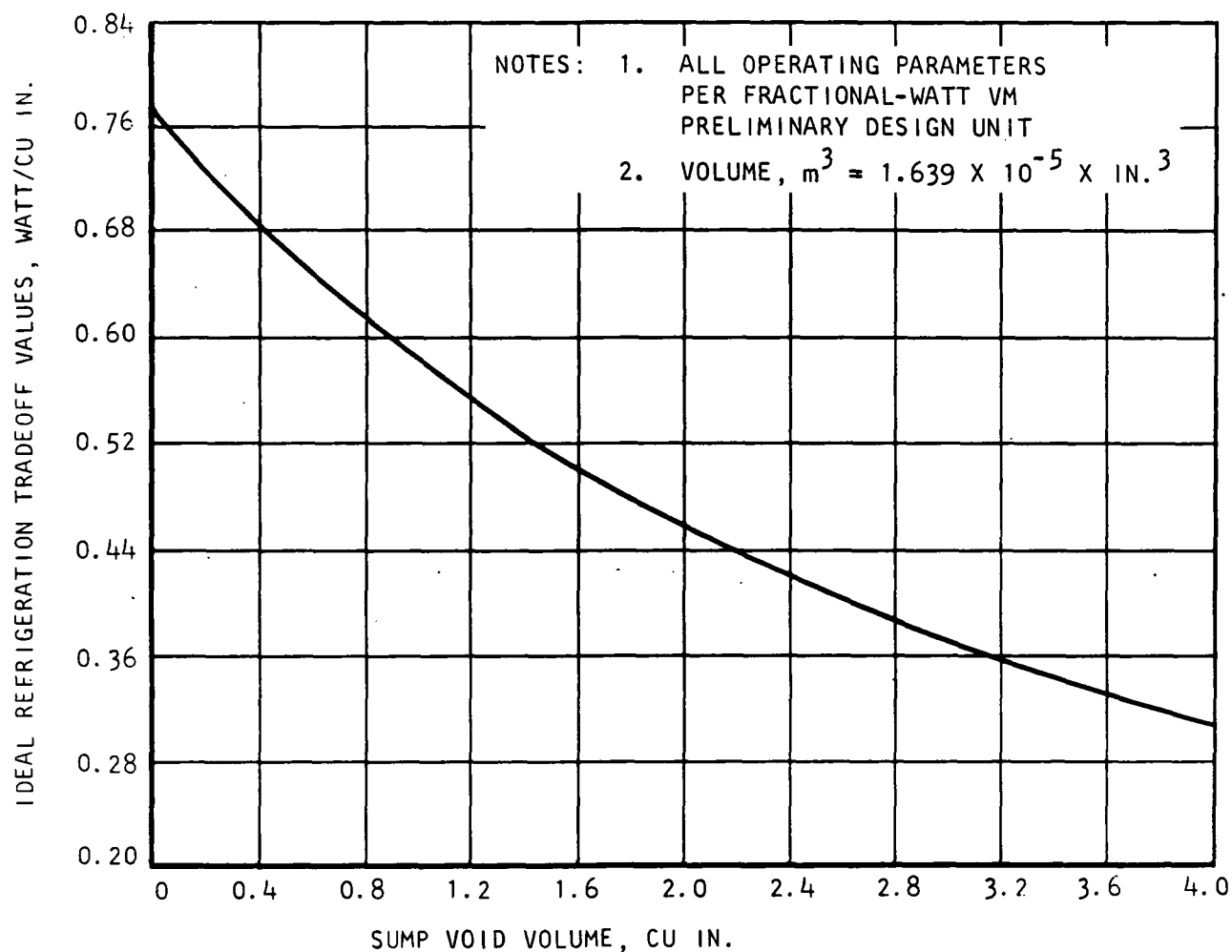


Figure 3-25. Effect of Cold Void Volume on Ideal Refrigeration Tradeoff Values



S-77549

Figure 3-26. Effect of Sump Void Volume on Ideal Refrigeration Tradeoff Values



is more representative of the mass in a given volume. Figure 3-27 presents the tradeoff factors of Table 3-15 as a function of ZT . A linear relationship is evident. Thus the tradeoff values could be reduced to a single constant for this particular machine, which would then be modified by a temperature-compressibility factor. The use of a plot such as Figure 3-27 is considered more simple when trade factors at various temperatures are desired.

TABLE 3-15

VOID VOLUME TRADEOFF PARAMETERS FOR PRELIMINARY DESIGN
FRACTIONAL WATT VM REFRIGERATOR

Location of Void Volume	Temperature of Void Volume, $^{\circ}K$ ($^{\circ}R$)	Tradeoff Factor $\Delta Q_c / \Delta V$, watts/ m^3 (watts/cu in.)
Cold end	62.3 (112)	1.143×10^5 (1.875)
Cold regenerator	204 (366)	0.380×10^5 (0.623)
Sump	344 (620)	0.232×10^5 (0.380)
Hot regenerator	559 (1077.5)	0.134×10^5 (0.2194)
Hot end	853 (1535)	0.935×10^5 (0.1531)

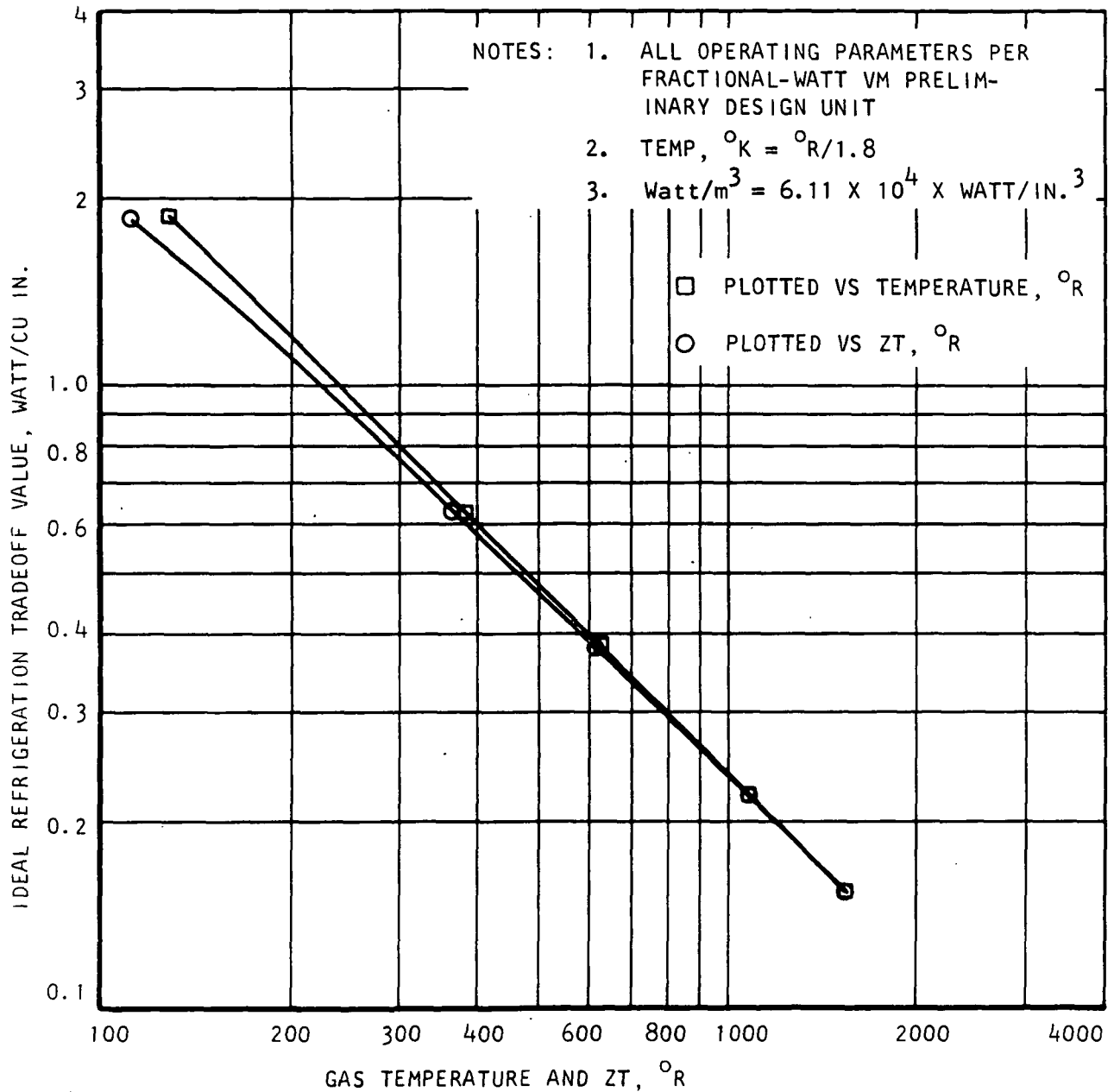
The effect of pressure drop between the sump and the cold end on ideal refrigeration was determined. This was done with the simplified approach described earlier under the computer program modifications. The pressure drop effect on ideal refrigeration is constant for maximum ΔP values from 6895 (1) to $3.79 \times 10^5 \text{ N/m}^2$ (55 psi) and the trade factor is $6.88 \times 10^{-6} \text{ watts/N/m}^2$ (0.04738 watts/psi). The corresponding increase in hot end heat input required is $2.66 \times 10^{-4} \text{ watts/N/m}^2$ (1.8356 watts/psi). Again, these values are specific to the preliminary design VM.

Flow Distribution Devices

1. Introduction

The prime importance of obtaining uniform flow distribution through all heat transfer devices, both heat exchangers and regenerators, has been repeatedly stressed throughout this report. Test results from the AiResearch IR&D VM refrigerator revealed non-uniform flow in the cold end of the refrigerator. The configuration of the cold end of the AiResearch VM refrigerator (cold end heat exchanger, displacer, and cold regenerator) is very similar to that of the GSFC VM refrigerator. Initial tests on the AiResearch refrigerator





S-77551

Figure 3-27. Refrigeration Tradeoff Values as a Function of Void Volume Temperature



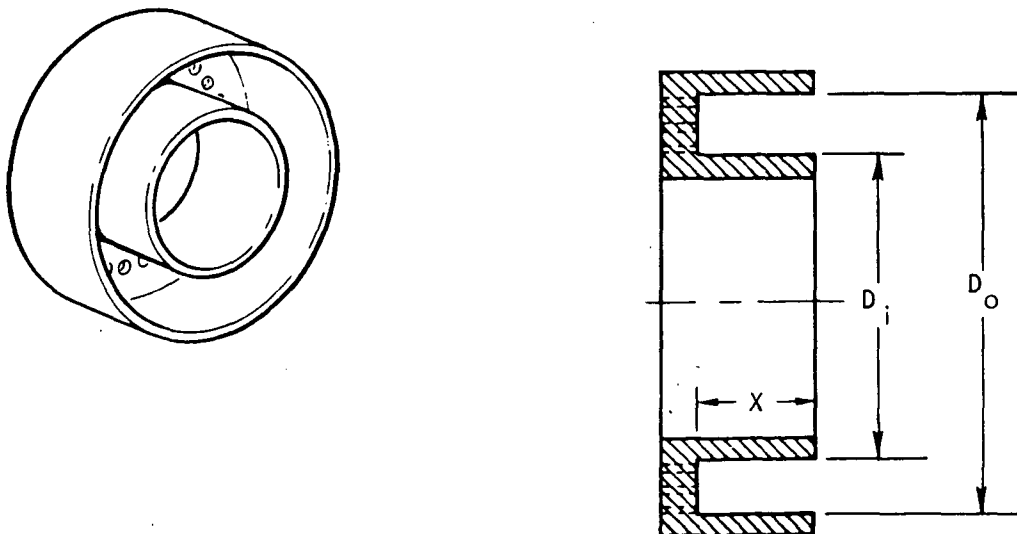
AIRESEARCH MANUFACTURING COMPANY
Torrance, California

indicated an unbalance in flow in the cold end heat exchanger and low temperature end of the cold regenerator. To overcome this problem, a flow distributor was designed and installed in this refrigerator. As a result of the successful testing of this flow distributor, the same basic design was incorporated into the GSFC 5 watt VM refrigerator. A similar device will be employed in the fractional watt machine. This device will be optimized by use of the refrigeration tradeoff factors developed in the previous section of this report. A generalized optimization procedure is described, and then applied to the cold end flow distributor. The procedure will be applied to other flow distribution devices within the machine as detailed design progresses.

2. Optimization Procedure

A flow distributor optimization procedure was developed utilizing the tradeoff factors determined in the preceding section of this report. Factors of prime importance affecting flow distributor design are the ratio of axial to circumferential pressure drop, the total pressure drop, and the void volume contribution to the machine. These factors are interrelated in such a manner that the combined effect on refrigeration is determined. This expression is then differentiated with respect to the axial dimension and set equal to zero. In this manner flow distributor dimensions are determined such that minimum loss of refrigeration occurs.

In general, the dimensions of a flow distributor, such as that at the end of a regenerator, are fixed by the geometry of the VM, with the exception of the axial length. A sketch of a typical flow distributor (Figure 3-28) illustrates this.



S-73895

Figure 3-28. Typical Flow Distributor

The void volume (ignoring that of the axial holes) may be expressed as a function of the variable axial length X:

$$V_D = \frac{\pi}{4} (D_o^2 - D_i^2) X = K_2 X \quad (3-19)$$

Thus,

$$K_2 = \frac{\pi}{4} (D_o^2 - D_i^2)$$

Assuming that the circumferential pressure loss is primarily shock (expansion, contraction, and turning of the gas) the pressure loss becomes a function of velocity head only. The velocity head varies as the square of the flow area.

$$A_C = (D_o - D_i) X \quad (3-20)$$

The axial pressure loss of a good flow distributor is generally 10 or more times the circumferential pressure loss. If this multiplying factor is denoted by N, the total pressure loss of the flow distributor is represented by Equation (3-21).

$$\Delta P = (N + 1) K_C \left(\frac{w}{A_C} \right)^2 \frac{1}{2g_C \rho} = \frac{K_1}{X^2} \quad (3-21)$$

where

K_C = the total loss coefficient in the circumferential direction

w = appropriate mass flow rate

g_C = gravitational constant

ρ = gas density

K_1 = is obtained by substituting Equation (3-20) and lumping all

$$\text{constant parameters together} = \frac{K_C w^2 (N + 1)}{2g_C \rho (D_o - D_i)^2} \quad (3-22)$$



The loss in refrigeration is expressed as the sum of the pressure drop contribution and the void volume contribution.

$$Q_L = C_1 \Delta P + C_2 V_D \quad (3-23)$$

where

Q_L = lost refrigeration

C_1 = pressure drop trade factor

C_2 = void volume trade factor

substituting Equations (3-19) and (3-21) in Equation (3-23), the loss in refrigeration due to the flow distributor is

$$Q_L = \frac{C_1 K_1}{X^2} + C_2 K_2 X \quad (3-24)$$

differentiating and setting the result equal to zero, the value of X is found to be:

$$X = \left(\frac{2 C_1 K_1}{C_2 K_2} \right)^{1/3} \quad (3-25)$$

This value of X represents the minimum loss of refrigeration due to the combined effects of pressure drop and void volume. The remainder of the design procedure consists of calculating the circumferential pressure drop, and multiplying it by N to determine the required axial pressure drop. The number and size of the axial holes is then selected to give this pressure drop and to comply with manufacturing techniques and flow distribution requirements.

This design method assures optimum flow distributors, and also applies to any component for which none of the design variables is prespecified. An example of its nonapplicability is the flow distribution slot between the two sections of the sump heat exchanger. For this slot, the axial pressure drop is fixed as that of the heat exchanger. Once the design value of N is selected, the dimensions of the slot are fixed and thus the void volume. The tradeoff for this particular slot involves engineering judgement as to what values of N are acceptable as a function of void volume.

An attempt was also made to apply this type of analysis to the hot and cold end heat exchangers, but the number of variables involved is too great. If the heat transfer conductance is considered as a design requirement, and thus a constant, some other variables such as hydraulic radius and ratio of friction factor to Colburn j modulus must also be specified. It is felt that this procedure would overly constrain the heat exchanger design. A single optimum design would not readily emerge from such an analysis.



3. Cold End Flow Distributor

The optimization procedure described above will now be applied to the cold end flow distributor.

The cold regenerator fixes the inner and outer diameter of the annulus into which the distributor is installed at 0.01078 m (0.424 in.) and 0.026 m (1.024 in.) respectively. Thus, allowing 0.000254 m (0.010 in.) thick walls on the distributor, $D_i = 0.01128$ m (0.444 in.) and $D_o = 0.0255$ m (1.004 in.) (See Figure 3-28 for nomenclature). Equation (3-19) thus yields $K_2 = 0.000411$ m² (0.638 in.²).

The axial to circumferential pressure drop ratio (N) is chosen as 15, and the total loss coefficient in the circumferential direction (K_c) is taken as 5.0. The maximum gas flow rate at the cold end of the cold regenerator (Table 3-7) is 0.000884 kg/sec (0.001943 lb/sec); this value is reduced by a factor of 12, since there are 12 flow slots in the heat exchanger. Substitution of these values into Equation (3-22) yields $K_1 = 0.00000234$ kgf (0.00000515 lbf). From the preceding section, the void volume trade factor at the cold end of the machine (C_2) is 1.143×10^5 watts/m³ (1.875 watts/in.³) and the pressure drop tradeoff factor (C_1) is 6.88×10^{-6} watts/N/m² (0.04738 watts/psi). Substitution of these values into Equation 3-25 yields

$$x = \left[\frac{2 C_1 K_1}{C_2 K_2} \right]^{1/3} = \left[\frac{2 \times 0.04738 \text{ watts} \times 0.00000515 \text{ lbf}}{1 \text{ lbf/in.}^2 \times 0.638 \text{ in.}^2 \times 1.875 \text{ watts/in.}^3} \right]^{1/3}$$

$$= 0.000188 \text{ m (0.0074 in.)}$$

Thus the optimum annular flow length for the cold flow distributor is slightly greater than 0.000179 m (0.007 in.). The total pressure drop for the flow distributor is obtained from Equation 3-21 as 648 N/m² (0.0940 psi) and the refrigeration loss due to this pressure drop is 4.45 milliwatts. The associated void volume is 7.73×10^{-8} m³ (0.00472 in.³) and the refrigeration loss is 8.85 mw. Thus the total refrigeration loss associated with the optimized flow distributor is approximately 13.3 mw.

The circumferential pressure drop is obtained by dividing the total pressure drop by (N + 1). Thus $P_c = 40.5$ N/m² (0.00587 psi) and the axial holes must be sized to yield a pressure drop of 607 N/m² (0.0881 psi). If fifty axial holes are chosen as a reasonable number from manufacturing considerations, and a loss coefficient of 1.5 velocity heads is assumed, the resulting hole size is 0.00034 m (0.0134 in.) diameter.



The flow distributor designed by this method yields minimum refrigeration loss. The axial holes are of a reasonable size, and the number of holes does not represent manufacturing problems. Other internal flow passages such as the sump access ports and ports from the sump heat exchanger to the ambient end of the hot displacer will be sized by this method as detail design of the VM progresses.

Design of Cold End Seal

The cold end seals function to control the rate of leakage which bypasses the cold regenerator. This function is critical in the refrigerator design since leakage past the seals can result in a significant loss in thermal performance. Leakage past the seals bypasses the regenerator by flowing through the annular space between the cold displacer and cold cylinder walls. At low leakage rates, the displacer and cylinder walls effectively regenerate the leakage fluid temperatures and the resulting thermal losses are small. As leakage rates increase, the walls can no longer function as an effective regenerator, and significant losses in overall thermal performance occur.

1. Design Configuration

The basic configuration of the cold end sealing system is shown in Figure 3-29. Seals incorporated in the cold end sealing system, from the cold end toward the sump end of the cold displacer, are as follows:

- (a) A 0.0254 m (1.0 in.) long, close-fit annular seal with a clearance between the inner wall of the regenerator and the seal of 0.0000635 m (0.0025 in.) maximum.
- (b) A 5-groove labyrinth seal with a tip clearance of 0.0000635 m (0.0025 in.), a groove spacing of 0.00127 m (0.05 in.), and a nominal tip width of 0.000127 m (0.005 in.)
- (c) The first linear bearing, which acts as an annular seal with a clearance of 0.00001017 m (0.0004 in.) and a length of 0.0254 m (1.0 in.).
- (d) The bearing support member, which acts as an annular seal with a clearance of 0.000089 m (0.0035 in.) and a length of 0.0429 m (1.69 in.).
- (e) The second linear bearing which is identical to the first.

The arrangement of the machine is such that these sealing elements are in series.



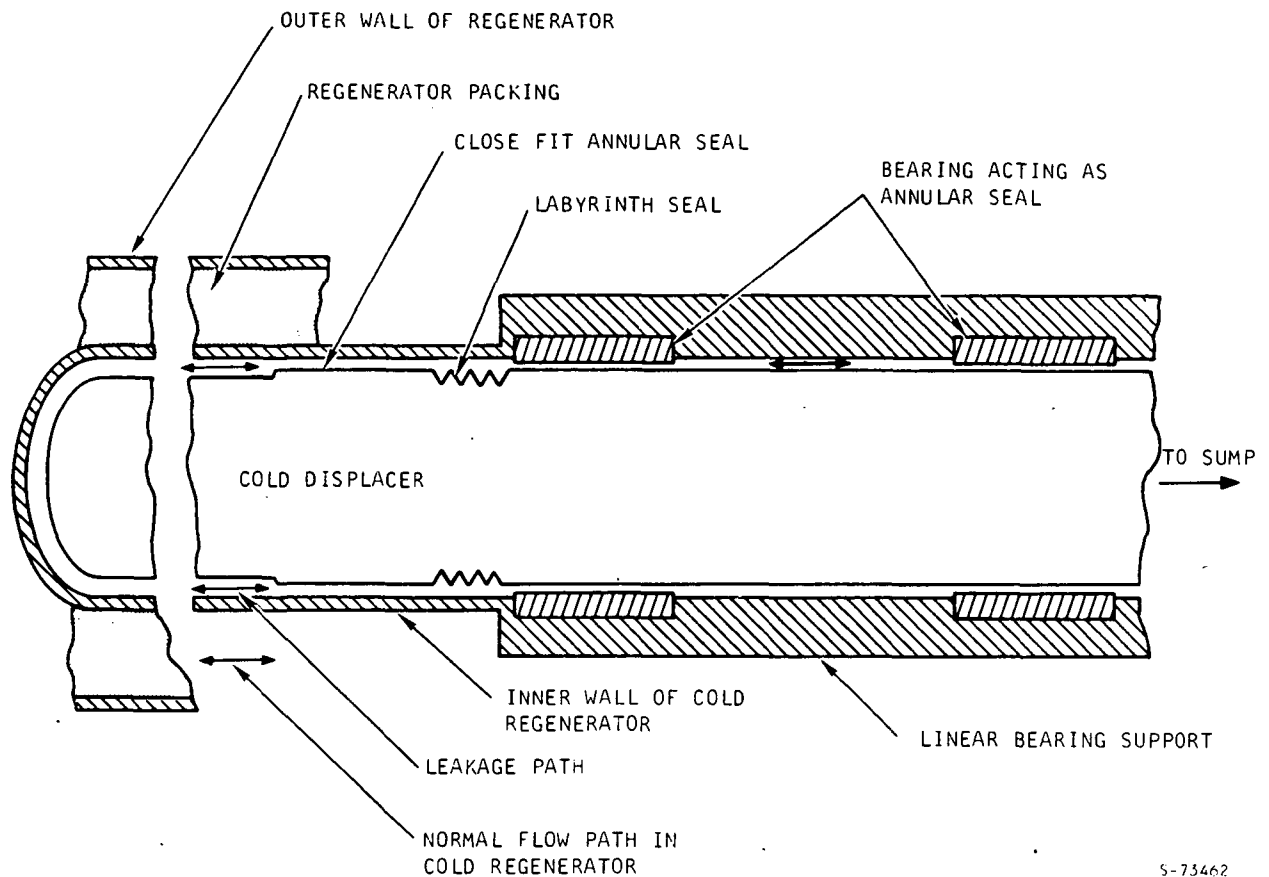


Figure 3-29. Cold Displacer Sealing Design Configuration

As with the GSFC 5 watt VM the linear bearings and the bearing support are used as part of the sealing system. The mechanical arrangement of the machine allows the use of these components as seals without penalty. The only disadvantage in their use is that the leakage rate is dependent on bearing clearance and increases as the bearing wears. However, the loading and rate of wear of these bearings is low, and will provide over two years of operation before bearing wear affects the performance (even in the worst case analysis).

2. Method of Analysis

The correlation for analyzing the leakage past both labyrinth and annular seals were developed and modified to match the seal test data in Ref. 3-1. The appropriate equations are:

for Labyrinth seal

$$\Delta P_i = \frac{\dot{\omega}_L^2 T_O R \eta}{C^2 A^2 P_O K_L^2 \zeta 2g_c} \quad (3-26)$$



for Annular Seal

$$\Delta P_i = \frac{\dot{\omega}_A L}{K_A \rho} \left[\frac{48\mu}{D_H^2 g_c A} \right] \quad (3-27)$$

Then noting that:

$$\Delta P_T = \sum_0^N \Delta P_i = \sum_0^N f_i(\dot{\omega}_i) \quad (3-28)$$

Where

P_T = the total pressure drop across the series of sealing elements

i and N = the identity and number of elements respectively

then further noting that

$$\dot{\omega}_T = \dot{\omega}_L = \dot{\omega}_A = \dot{\omega}_i \quad (3-29)$$

that is, the flow past each element is the same, the pressure drop can be computed as a function of leakage rate from Equation (3-28).

3. Performance Characteristics

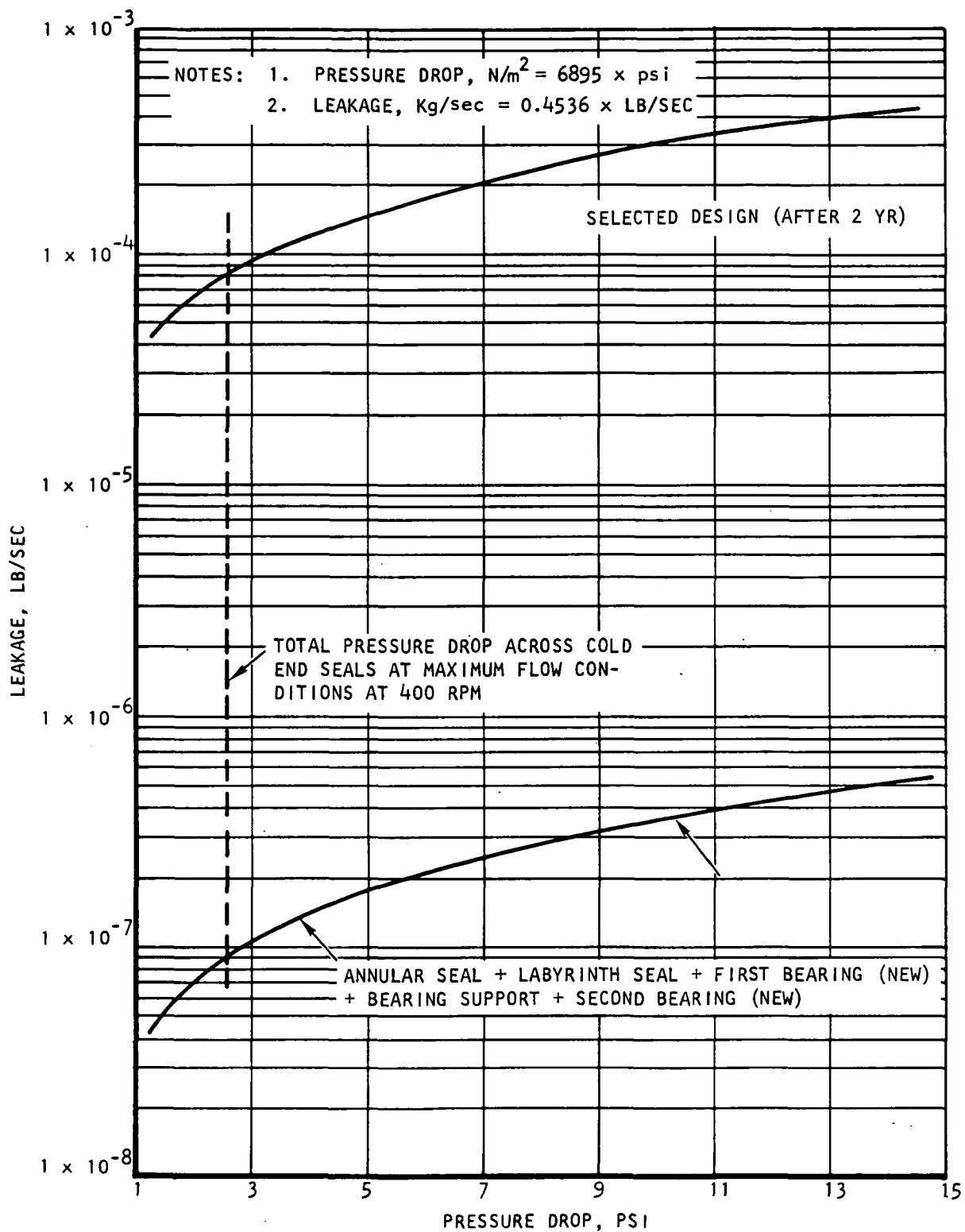
a. Leakage Rate

The configuration of the cold end of the refrigerator is such that seal elements can be added in series (or subtracted) as desired by providing flow passages into the active cycle volume at different locations. Figure 3-30 gives the leakage rate as a function of pressure drop for the combination of all seal elements in series. The design combines all possible seal elements to minimize the losses due to leakage.

Figure 3-30 presents leakage rate data for seal systems with both new bearings and bearings after 2 years of wear. As the bearings wear, the clearance of the annular seal (the seal which the bearings form with the cold displacer) increases and hence the leakage rate increases. The bearing wear rate used in the analysis is based on the bearing material tests described in reference 3-1. The use of these wear rate data is believed very conservative due to the low loading of the linear bearings compared to the wear rate test conditions.

The dashed vertical line in Figure 3-30 corresponds to the preliminary design limit level of pressure drop across the cold end seal. This pressure drop level is based on the maximum cold end flow rate at a rotational speed of 400 rpm and thus represents the worst case conditions.





S-77559

Figure 3-30. Cold-End Seal Leakage--Pressure Drop Characteristics



AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California

72-8846
Page 3-85

b. Thermal Losses

The cold end leakage is important, in that it influences refrigerator thermal performance. A good approximation of the thermal losses associated with the cold end leakage is provided by the simple model developed below.

Figure 3-31 depicts the basic elements of the thermal loss model employed. Taking an element of length (dx) along the annular flow passage between displacer and inner regenerator wall, the following differential equation can be written:

$$\frac{dT_f}{dx} + \frac{hA_c}{\dot{w}C_p} T_f = \frac{hA_c}{\dot{w}C_p} T_w \quad (3-30)$$

Where

T_f = temperature of the leakage gas at any location X

T_w = temperature of the regenerator and displacer walls at any location X

h = local heat transfer coefficient between leakage gas and the surrounding walls

A_c = heat transfer area per unit length along the leakage path

$\dot{w}$ = rate of leakage

C_p = heat capacity of leakage gas

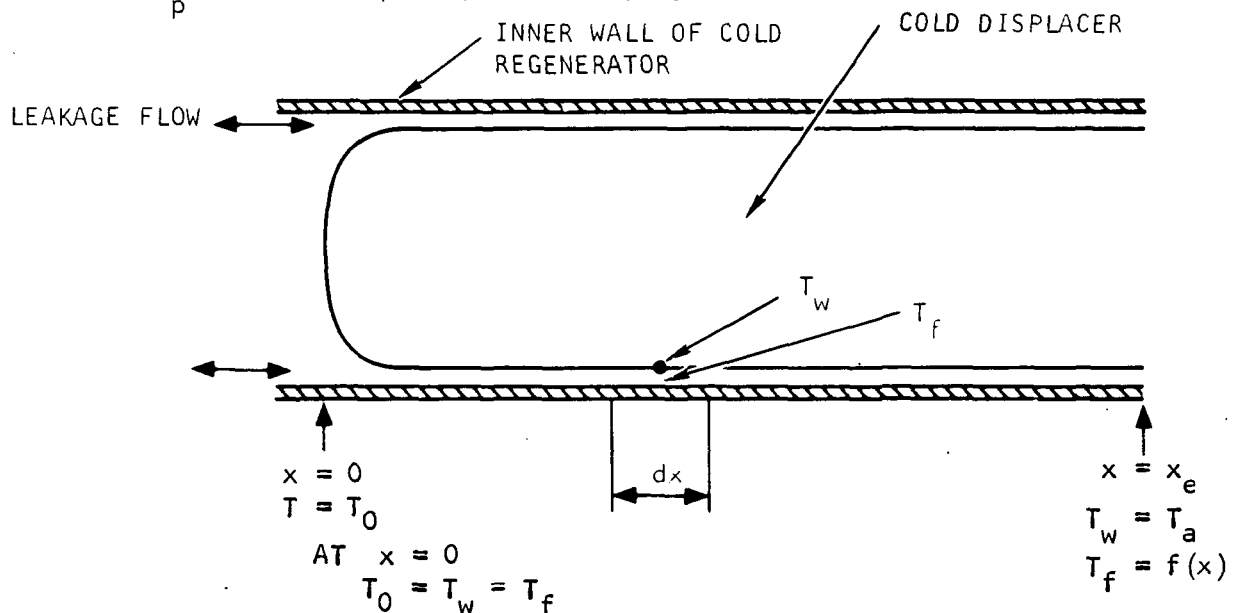


Figure 3-31. Leakage Thermal Loss Model

Then if a linear temperature distribution is assumed along the displacer and regenerator walls and ratio of heat transfer coefficient to gas heat capacity is taken as a constant, Equation 3-30 can be written as:

$$\frac{dT_f}{dX} + \alpha T_f = \alpha (T_a - T_o) \frac{X}{X_e} + \alpha T_o \quad (3-31)$$

Where the new terms are:

$$\alpha = \frac{hA_c}{\dot{w}C_p}$$

T_a = wall temperature at the sump end of the displacer

T_o = refrigeration temperature or wall temperature at $X = 0$

X_e = length of the displacer

Solving Equation 3-31 with the boundary conditions of $T_f = T_o$ at $X = 0$ yields:

$$T_f = \frac{T_a - T_o}{X_e} \left\{ X - \frac{1}{\alpha} + \frac{1}{\alpha} e^{-\alpha X} \right\} + T_o \quad (3-32)$$

If the leakage flow were completely regenerated by the walls, the temperature of the fluid would be that of the wall at the end of the displacer; that is:

$$T_f \longrightarrow T_a @ X = X_e$$

or if flow (leakage) is considered in the reverse direction

$$T_f \longrightarrow T_o @ X = 0$$

In setting up the relation for T_f , this latter condition was used as a boundary condition assuming flow in the positive X direction (see Figure 3-31). Relationships for flow in the reverse direction are similar but are not developed here. In the actual case, due to the cyclic operation of the VM refrigerator, the leakage flow does reverse direction. The leakage losses can be estimated, however, by considering flow in one direction with an appropriate time span. With this consideration the thermal losses per cycle due to leakage can be expressed as:

$$Q_L = \oint |\dot{w}| C_p \left\{ (T_a - T_o) \frac{1}{\alpha X_e} (1 - e^{-\alpha X_e}) \right\} d\tau \quad (3-33)$$



Where

$\oint$ implies integration around the cycle and

τ = time

Assuming a constant leakage rate, the leakage thermal losses can be expressed as:

$$\dot{Q}_L = \dot{m} C_p \left\{ (T_a - T_o) \frac{1}{\alpha X_e} (1 - e^{-\alpha X_e}) \right\} \quad (3-34)$$

Figure 3-32 gives the losses (Equation 3-34) as a function of leakage rate. The thermal losses at leakage rates corresponding to the design limit levels of pressure drop and associated cold end leakage rates (see Figure 3-30) are shown in Figure 3-32. It is noted these losses are estimated for the machine after two years of wear; thermal losses due to leakage are negligible for a new machine. In fact, the leakage thermal losses for the machine after 2 years of wear are expected to be considerably lower than indicated in Figure 3-32 for two reasons. First, the bearing wear rate used in estimating the leakage is considered to yield a conservative result as previously mentioned. Second, constant leakage rates at their maximum levels were used in estimating the losses by use of Equation 3-33. In the actual case, the leakage rate is a periodic function with an average value on the order of 70 percent of the maximum. Thus not only are the losses lower due to the decreased leakage flow, but the regenerator and displacer walls more effectively regenerate the leakage gas, further reducing the losses below the maximum values indicated in Figure 3-32.

Design of Hot End Seal

The hot-end seal functions to control the leakage rate of the working fluid which bypasses the hot regenerator. Leakage bypasses the regenerator by flowing through the annular space between the hot displacer and the inner wall of the regenerator. Excessive leakage results in a loss in thermal performance, therefore hot end seal design is an important consideration.

At low leakage rates, the hot displacer and cylinder walls effectively regenerate the leakage fluid temperatures and the resulting thermal losses are small. As the leakage rates increase, the walls can no longer function as an effective regenerator; thus significant losses in overall thermal performance result.

1. Design Configuration

The hot end seal configuration (Figure 3-33) consists of the following two elements, from the hot end toward the sump end: (1) a 5-groove labyrinth seal with a tip clearance of 0.0000635 m (0.0025 in.), a groove spacing of 0.00127 m (0.05 in.), and a nominal tip width of 0.000127 m (0.005 in.) and (2) a 0.0254 m (1.0 in.) long close fit annular seal with clearance between the inner wall of the regenerator and the seal of 0.0000635 m (0.0025 in.) maximum.



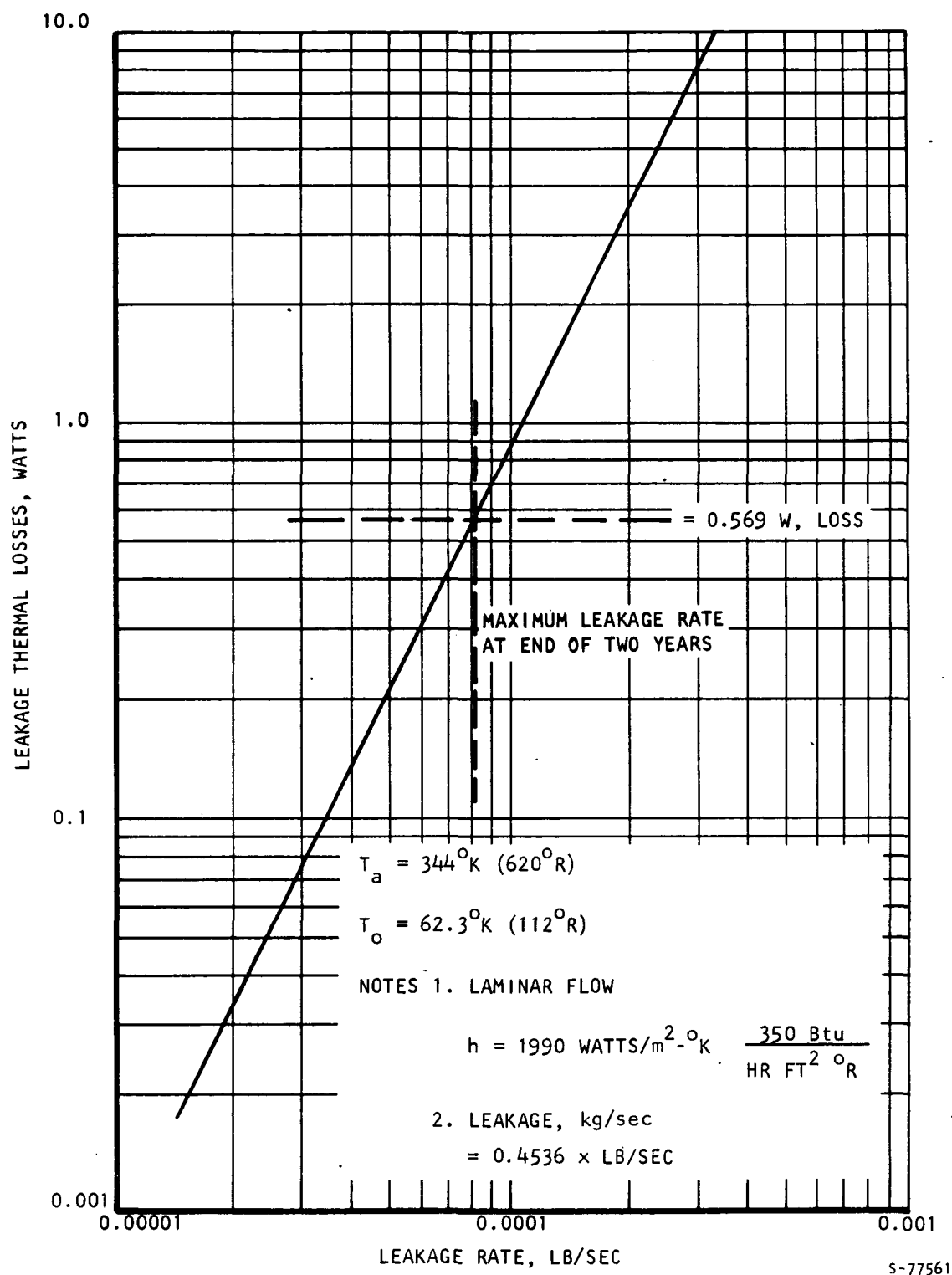


Figure 3-32. Cold End Thermal Losses Due to Leakage as a Function of Leakage Rate



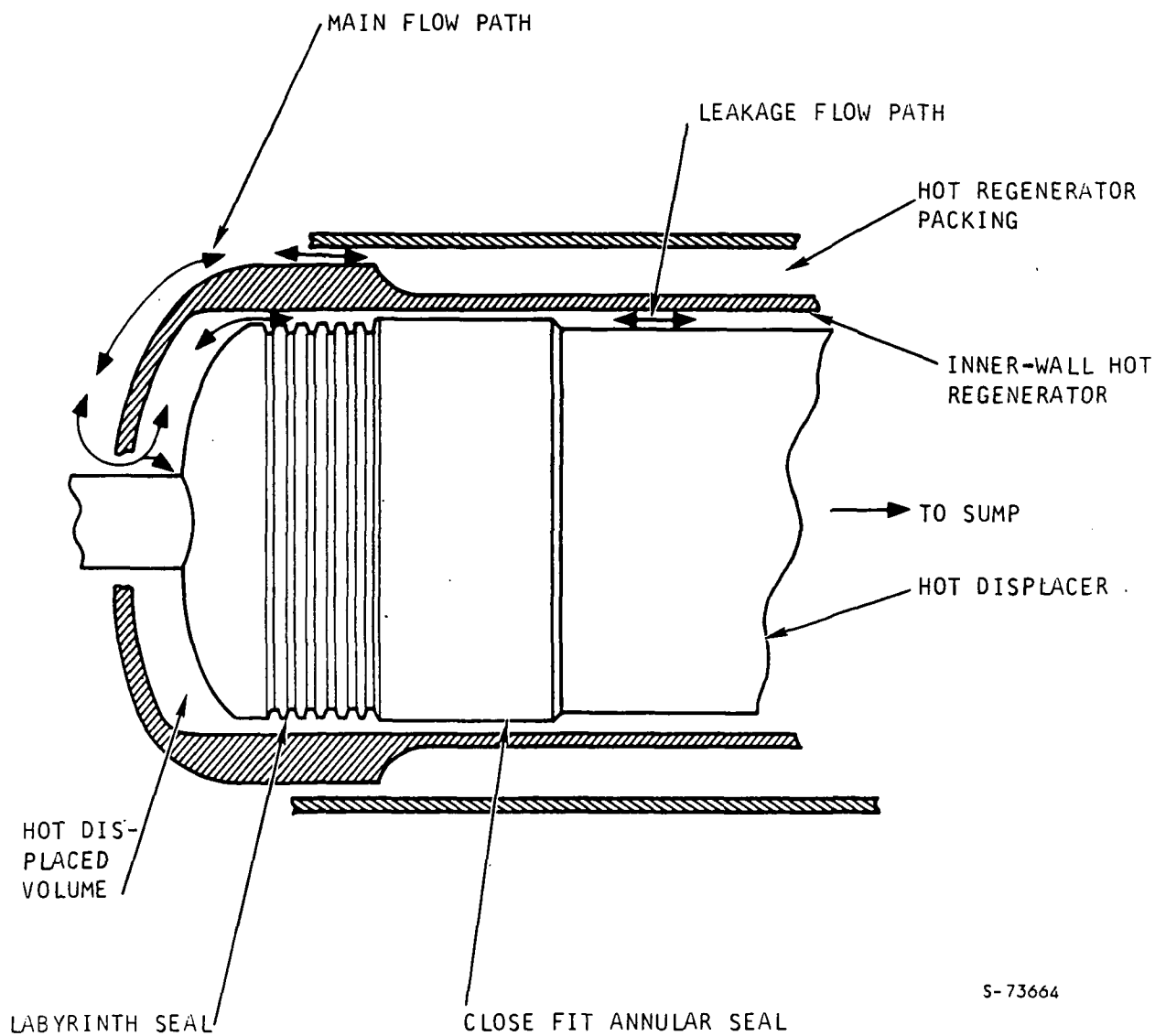


Figure 3-33. Hot-End Seal Configuration



These sealing elements are similar to those used in the cold-end seal except for the diameter. One major difference is the use of linear bearings as part of the sealing system for the cold end. The bearings are not used in the hot-end seal and thus the performance of this seal is independent of wear (or operational time).

It should be noted that the seal is located in the highest temperature region possible within the machine. The leakage rate is an inverse function of the temperature, therefore, this location minimizes the leakage rate for a given seal configuration. The seal design selected provides very low losses as discussed in the following paragraphs.

2. Method of Analysis

The method of analysis is identical to that used for the cold end seals.

3. Performance Characteristics

Figure 3-34 gives the hot-end seal leakage rate as a function of the pressure drop across the seal. The design limit pressure drop and the associated leakage rate shown in Figure 3-34 corresponds to the preliminary design goal maximum pressure drop across the seal. This pressure drop includes the maximum pressure drops across: (1) sump ports to backside of hot displacer, (2) Section 1 of the sump heat exchanger, (3) interface between sump heat exchanger and hot regenerator, (4) hot regenerator and (5) the hot-end heat exchanger. Due to the use of maximum pressure drops--pressure drops corresponding to the maximum flow rates at a rotational speed of 400 rpm--the indicated design leakage rate is conservative.

Comparing Figure 3-34 with Figure 3-30 (Figure 3-30 gives the cold-end leakage rate vs pressure drop) a major difference is noted. The hot-end seal is completely non-contacting and is not subject to wear. The hot-end leakage is therefore independent of operational life.

Figure 3-35 gives the thermal loss for leakage past the hot-end seal as a function of leakage rate. The method of calculating this thermal loss, which takes into account the regenerative effect of the displacer and cylinder walls, is identical to that used for the cold-end seal thermal losses. The loss indicated for the design limit conditions is not significant compared to the 80 watts maximum power input to the hot-end required to operate the system.

HEAT PIPE INTERFACE

This section summarizes the effort expended on Subtask 1.3 entitled Heat Pipe Interface Definition. The objective of this effort is to evaluate the design of a removable heat pipe interface assembly, connecting the VM sump to a space radiator, and to carry the design only to a depth necessary to define the interface requirements.

The design approach is the same as that used in the GSFC 5 watt VM effort, and it employs an interface clamp design which is adaptable to use with either



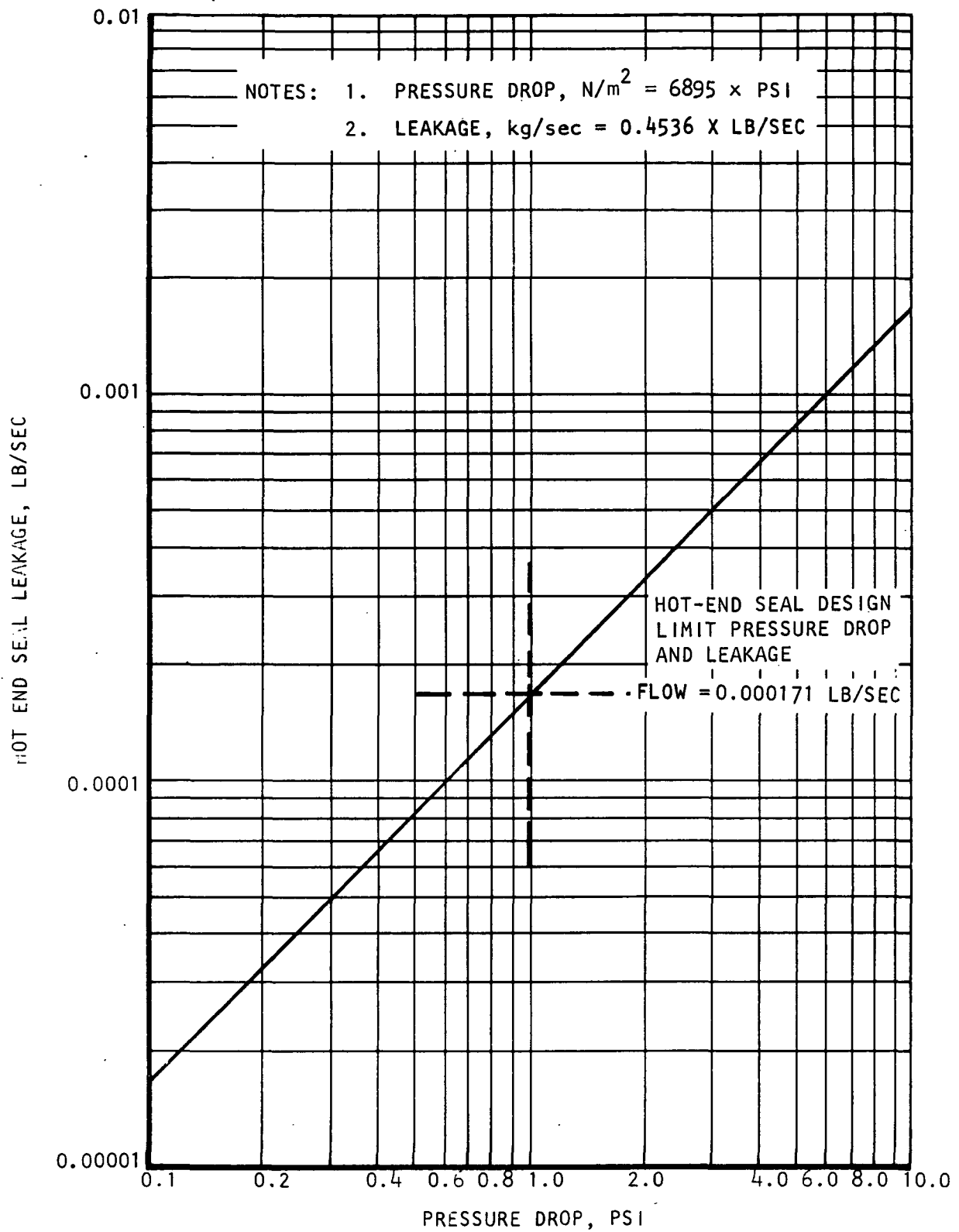
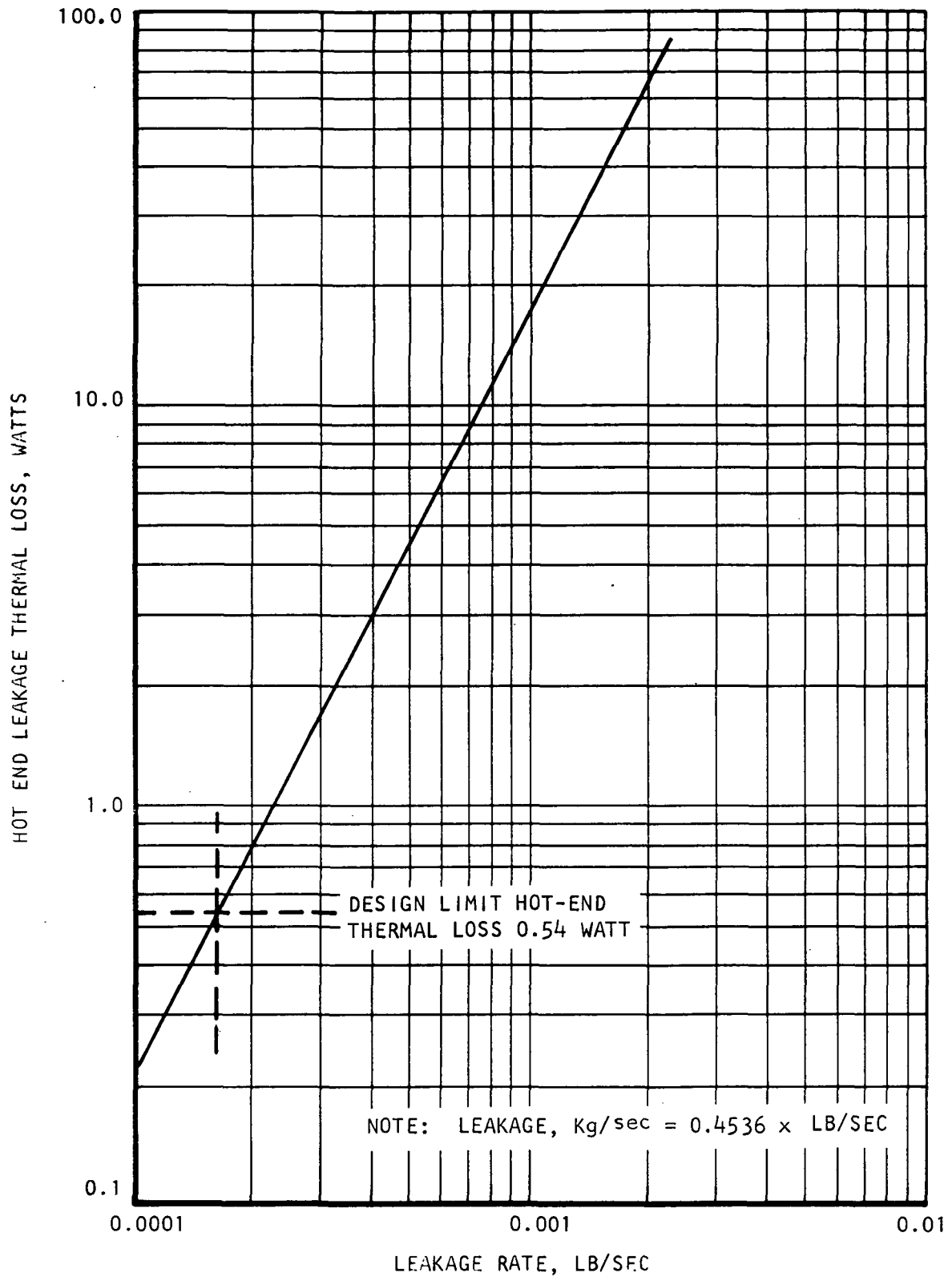


Figure 3-34. Hot-End Seal Leakage Rate vs Pressure Drop

S-77560





S-77558

Figure 3-35. Hot-End Thermal Loss vs Leakage Rate



a water cooling system or an ambient heat pipe assembly. The latter consists of one or more heat pipes on each side of the refrigerator crankcase designed to allow for the redundancy requirement.

Design Requirements

The system parameters influencing the design of the heat pipe assembly are listed below.

Heat rejection (including motor losses)	70 w
Redundancy requirement	50-percent
Sump diameter	0.0635 m (2.5-in.)
Sump length (allocated for heat pipe assembly)	0.0556 m (2.1875-in.)
Maximum cycle pressure	$6.895 \times 10^6 \text{ N/m}^2$ (1000 psia)
Minimum cycle pressure	$6.10 \times 10^6 \text{ N/m}^2$ (885 psia)
Sump temperature	334°K (140°F)

The present effort will consist of the following items

- (a) Sink temperature selection
- (b) Heat pipe design
- (c) Heat pipe assembly design
- (d) Interface definition
- (e) Interface temperature drop

Sink Temperature Selection

The sink temperature in this case refers to the spacecraft radiator temperature. The lower the sink temperature the higher the available temperature drop between the sump of the refrigerator and the radiator, which simplifies the design of the heat pipe assembly. On the other hand, the lower the sink temperature the larger the radiator area which is required to reject a given heat load.

A NASA analysis* of the heat rejection radiator was used in conducting system tradeoffs to select the sink temperature for the heat pipe assembly. The analysis is based on the following assumptions.

*By Dennis J. Asato, NASA, Goddard Space Flight Center.



- (1) Isothermal radiator surface
- (2) Environmental factor (ξ): 0 to 0.6
- (3) $\xi = f\left(\frac{\alpha}{\epsilon}, \text{spacecraft orientation}\right)$

Variables in the analysis are defined below

A = energy absorbed by radiator

α = absorptivity

E = energy emitted by radiator

ϵ = emissivity

$$\xi = \frac{(\text{energy absorbed})}{(\text{energy emitted by isothermal radiator})}$$

P = power, watts

S = surface area, ft^2

$$\sigma = 53.2858 \times 10^{-10} \frac{\text{watts}}{\text{ft}^2 \text{ } ^\circ\text{K}^4}$$

T = temperature, $^\circ\text{K}$

The relationship between the radiator area and the radiator temperature is obtained as follows

$$\xi = \frac{A}{E} \quad (3-35)$$

$$E = P + A \quad (3-36)$$

$$E = \sigma \epsilon T^4 S \quad (3-37)$$

Eliminating A from Equations (3-35) and (3-36) one gets:

$$P = E (1 - \xi) \quad (3-38)$$

Substituting Equation (3-37) into Equation (3-38):

$$P = (1 - \xi) \sigma \epsilon T^4 S \quad (3-39)$$

$$\boxed{\frac{S\epsilon}{P} = \frac{1}{(1 - \xi) \sigma T^4}} \quad (3-40)$$



The radiator surface area versus the environmental factor ξ is shown in Figure 3-36 for various radiator temperatures. Figure 3-37 depicts the radiator area versus the radiator temperature for two values of ξ (0.3 and 0.45). A decrease in the radiator temperature from 322 (120) to 316.5°K (110°F) results in an increase of about 6-percent in the radiator surface area. Hence, a preliminary choice of sink temperature range of 316.5 (110) to 322°K (120°F) appears to be satisfactory. Hence, the available temperature drop for the heat pipe assembly ranges from 4.1 (20) to 16.7°K (30°F).

Heat Pipe Design

Design problems such as working fluid selection, material selection, wick design, selection of a heat pipe diameter, power transport capability, temperature drop, etc., are discussed in the following paragraphs.

1. Working Fluid Selection

There are four basic criteria that govern the selection of a working fluid for the heat pipe. These criteria that vary in relative importance with the particular application are:

- Large transport factor, $N_\ell = \frac{\rho_\ell \sigma h_{fg}}{\mu_\ell}$
- High thermal conductivity
- Moderate vapor pressure
- Chemical compatibility with wick and container

The transport factor N_ℓ must be chosen as large as possible for the operating temperature range. Figure 3-38 presents the transport factor as a function of temperature for several working fluids. Water is shown for comparison purposes only; it is not a candidate fluid because of its high freezing point. It is clear that ammonia is the most attractive on the basis of transport factor since N_ℓ is from 1.3 to 10 times that of its closest competitor, methanol.

It is also desirable to have a working fluid with a high thermal conductivity, which reduces the conduction temperature drop through the wick. Thermal conductivities of candidate working fluids for the present application are shown in Figure 3-39. Again, ammonia is shown to be superior.

The vapor pressure of the working fluid should not be so high that the heat pipe must become a pressure vessel with prohibitively thick walls. It should also not be so low that the vapor densities result in high-pressure drops in the moving vapor. The vapor pressure-temperature curves for the two fluids discussed above are shown in Figure 3-40. Ammonia has the highest saturation pressure. Its high vapor pressure at the temperature level of



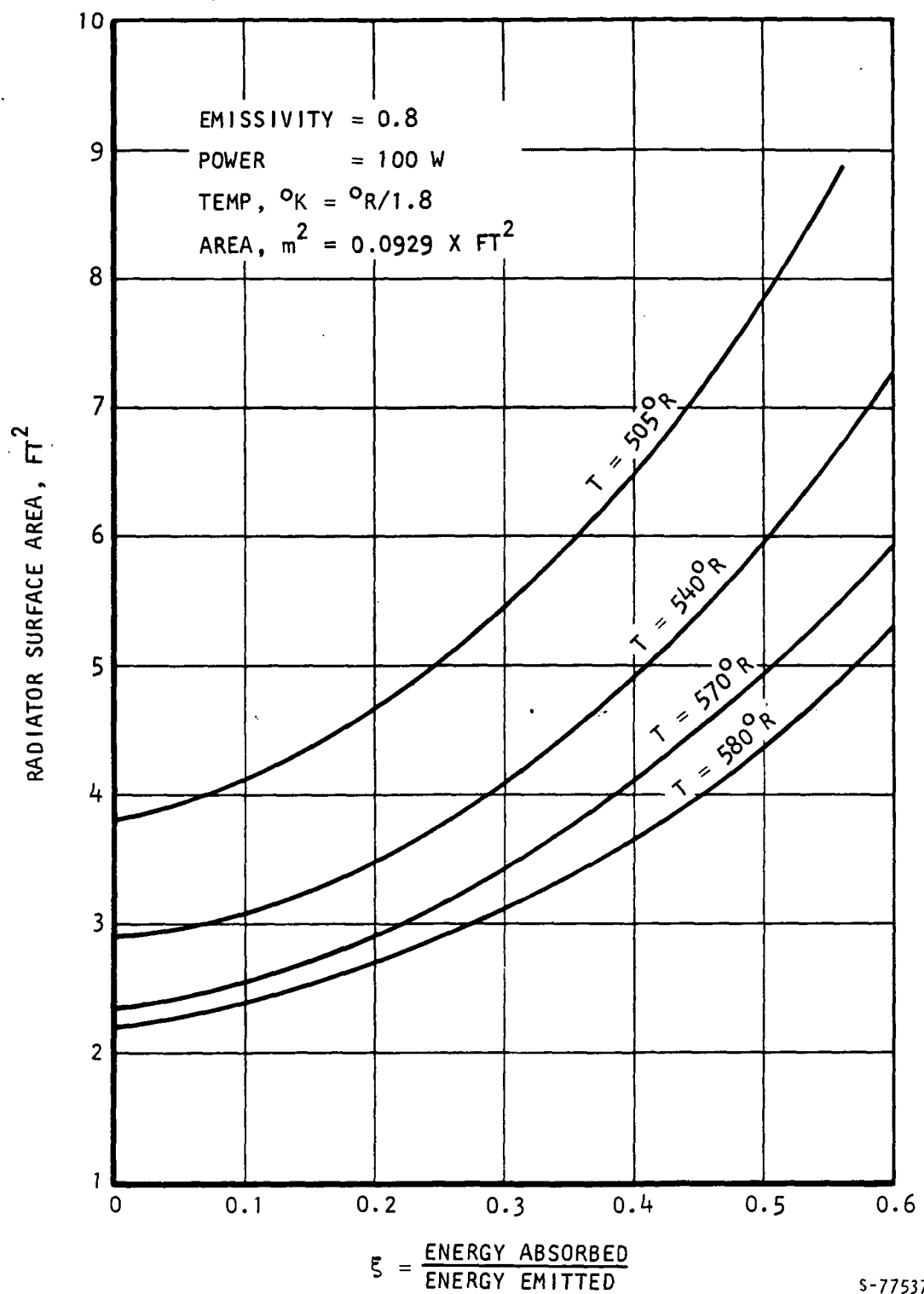


Figure 3-36. Space Radiator Design Chart



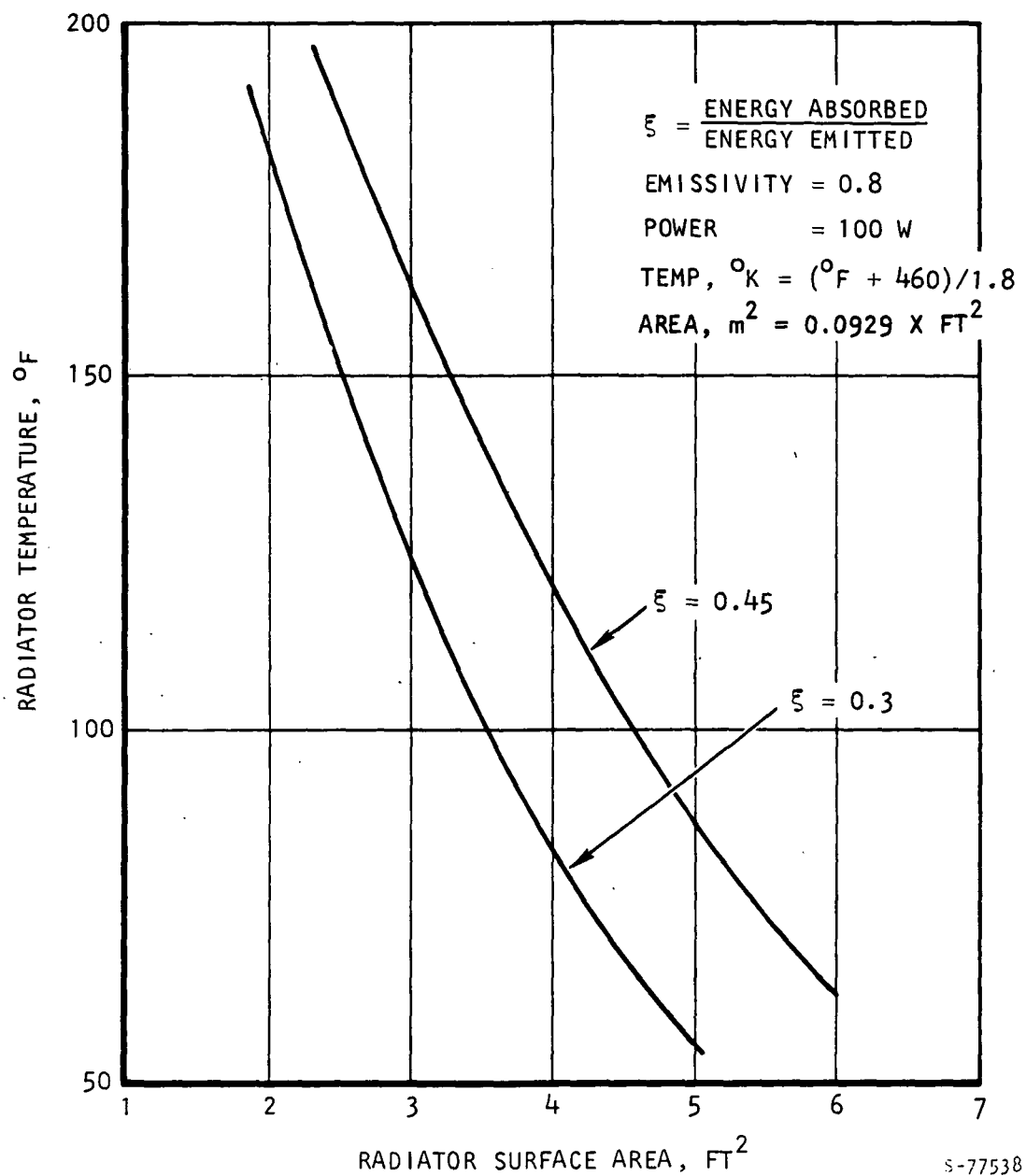
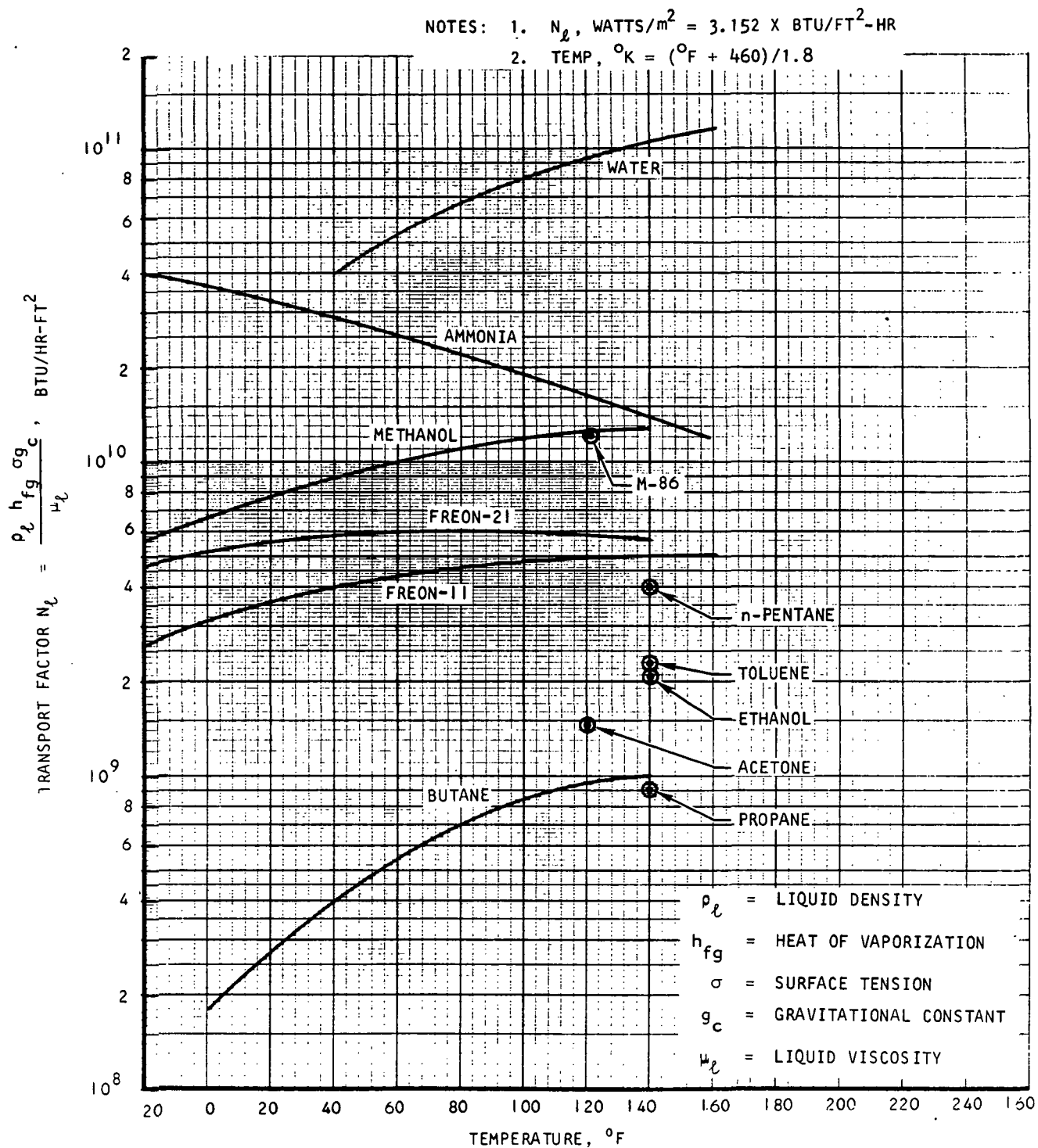


Figure 3-37. Effect of Radiator Area On Radiator Temperature

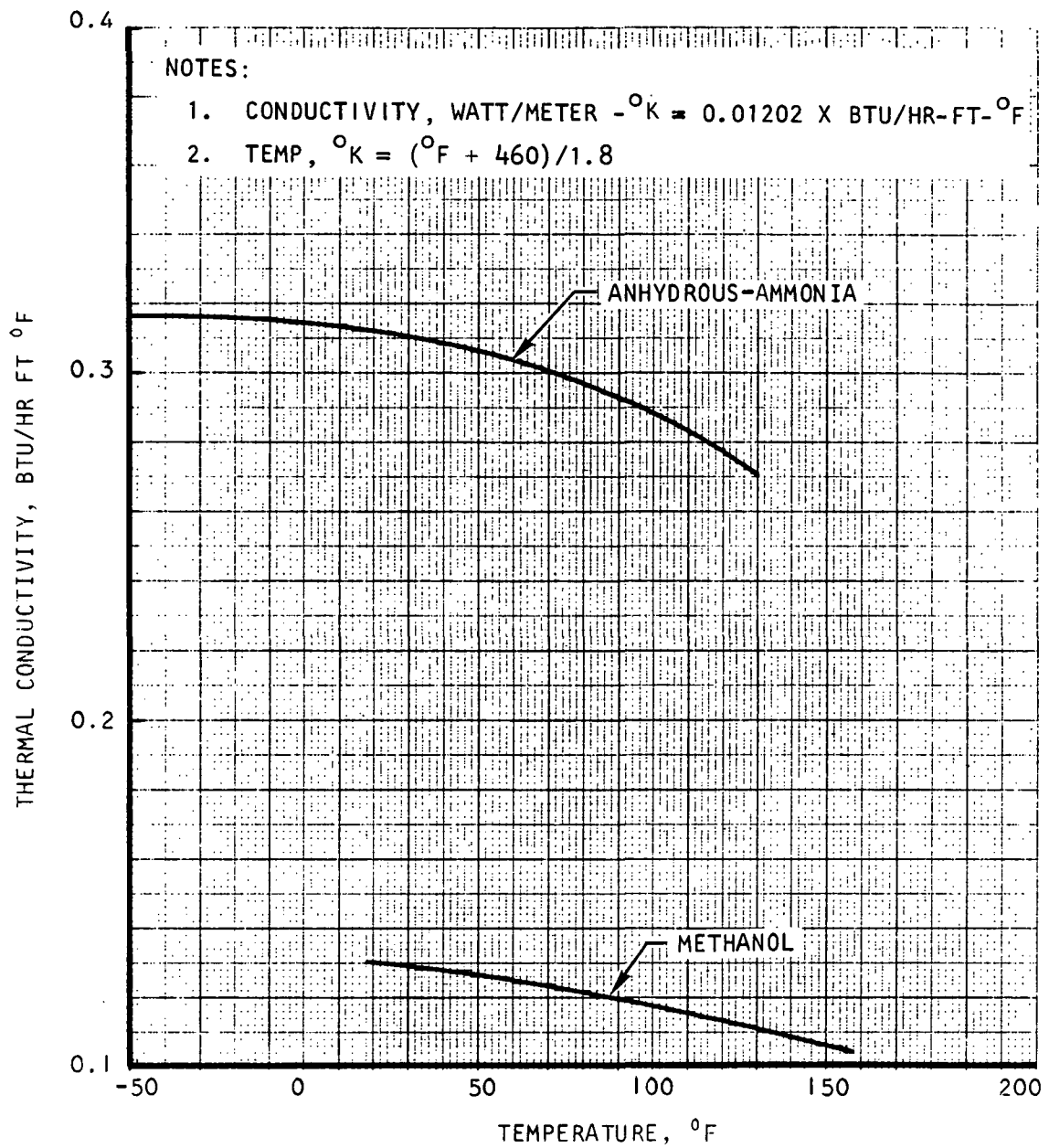




S-63640 -A

Figure 3-38. Transport Factors of Various Heat Pipe Working Fluids





S-63639 -A

Figure 3-39. Thermal Conductivity of Candidate Fluids



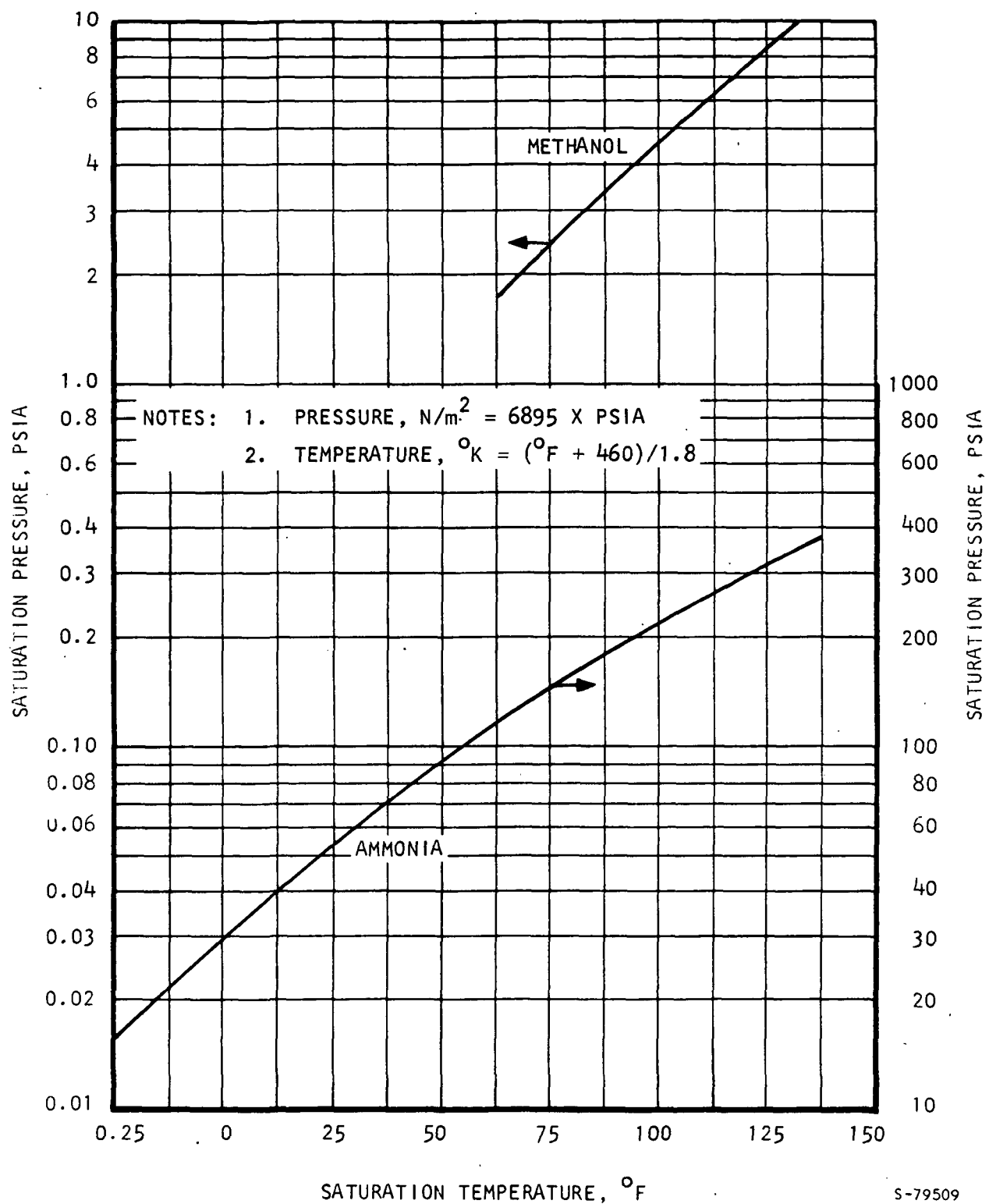


Figure 3-40. Pressure-Temperature at Saturation



interest will yield low vapor pressure drops with corresponding small saturation temperature drops between evaporator and the condenser. Stated another way, it will allow small vapor passage diameters without increasing the vapor pressure drop. The vapor pressure is also low enough to allow heat pipe construction utilizing reasonable material gages. The vapor pressure of methanol is too low for the present application.

Finally, there must be chemical compatibility between the working fluid, the wick, and the container.

2. Material Selection

Aluminum is a very reactive metal. Its resistance to corrosion and chemical attack is due to a thin film of aluminum oxide (Al_2O_3) that forms on the surface immediately upon exposure to the atmosphere. The corrosion resistance of aluminum to materials other than water is generally very good and has resulted in the use of aluminum in the handling of many substances. This reactive metal depends upon its alumina surface film for its resistance to chemical attack, and any material that degrades or removes this film may cause increased corrosion.

In contrast to most corrosion problems, in the heat pipe application, the structural integrity of the tube wall is not the primary consideration. Rather, it is the quantity of noncondensable gas generated in the thermal decomposition or corrosion reaction that must be avoided because such gas collects in the condensing region and effectively prevents the flow of working fluid vapor into this region.

While the compatibility of ammonia with aluminum has been demonstrated, considerable care must be taken to assure that the ammonia is pure. Purities less than 99.99 percent have been found to be corrosive in heat pipes, particularly if the impurities contain water vapor.

On the basis of the above discussion, it appears that ammonia-aluminum is the best combination for the present application.

3. Wick Design

The desirable wick characteristics are dictated by the two functions served by the wick, i.e., to deliver and distribute liquid to the evaporator region and to serve as a heat transfer path between the heat pipe container and the liquid-vapor interface where phase change occurs. The ideal wick configuration will have:

- Low resistance to liquid flow
- High capillary pumping force
- Low ΔT at evaporator and condenser
- Insensitivity to nucleate boiling



For a homogeneous wick, properties that enhance one of the above characteristics will penalize another. For instance, low resistance to fluid flow means large pores while high capillary pumping means small pores or using a small pore wick; low resistance to fluid flow means a thick wick while low ΔT means a thin wick. In the case of a homogeneous wick, therefore, selection of the wick configuration involves a tradeoff between the various properties.

One method of obtaining wick configurations that possess all of the desirable characteristics to a greater extent has been the use of composite wicks. Several configurations have been proposed, and some of these (channels, multilayer screens of different mesh sizes, and I-beam wicks) have been evaluated experimentally.

Additional factors influencing wick design are chemical compatibility with the working fluid and the container (in this application, aluminum), and separation of the liquid and vapor flow streams.

Two candidate wick design concepts are shown in Figure 3-41. In Concept 1, a composite wick is made of two layers of screen of different mesh sizes. The loose mesh is used for the liquid return from the condenser to the evaporator end to reduce viscous losses in the liquid path. The fine mesh screen is used for capillary pumping and to separate the liquid from the high velocity counter-current vapor flow. A retaining spring is used to maintain the screen in contact with the tube wall in applications which require bending of the heat pipe.

In Concept 2, the main liquid path is isolated from the heat path, which usually results in a smaller temperature drop in the heat pipe. Again in this concept, the loose mesh screen is used for the liquid path while the fine mesh screen provides the capillary pumping. At the evaporator end liquid is delivered to the tube wall through tabs on both ends, and the liquid is distributed around the tube through the narrow circumferential grooves.

While Concept 2 provides lower temperature drop in a heat pipe, Concept 1 offers more structural integrity, particularly during bending of the heat pipe. The advantage of smaller temperature drop is of negligible importance in the present application due to the low level of the power transported. At these levels, the differences in ΔT between the two concepts are quite small.

Since Concept 1 was used in the ambient heat pipe for the 5-watt VM machine, and since experimental data are available for these heat pipes, it was decided to use the same concept for the heat pipes on the fractional watt VM machine.

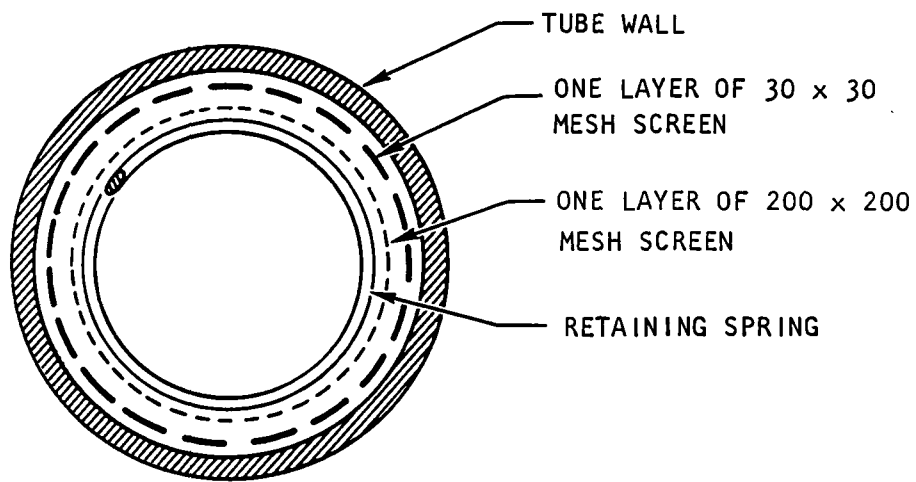
4. Power Transport Capability

The design of a heat pipe to meet the specified set of conditions starts with satisfying the pressure drop criterion:

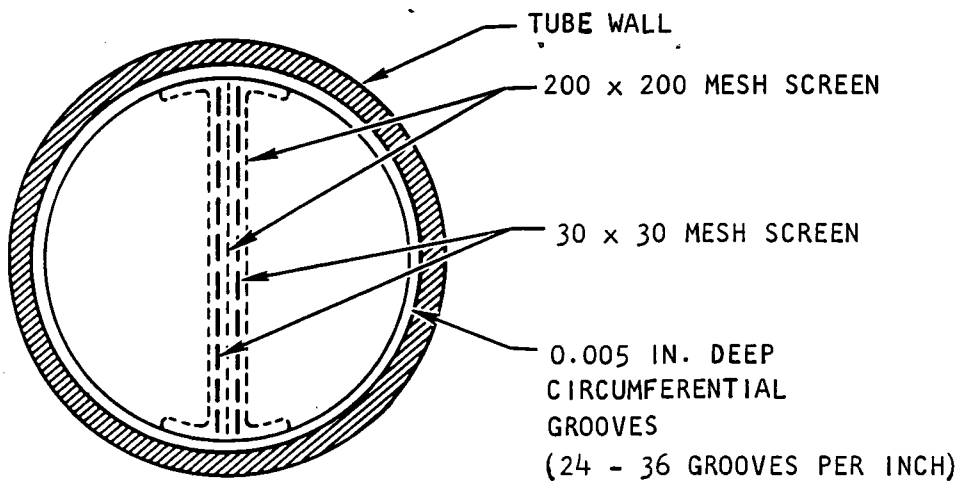
$$\Delta P_c \geq \Delta P_l + \Delta P_v \pm \Delta P_g$$

$$\left[\begin{array}{c} \text{net capillary} \\ \text{head} \end{array} \right] \geq \left[\begin{array}{c} \text{liquid pressure} \\ \text{drop} \end{array} \right] + \left[\begin{array}{c} \text{vapor pressure} \\ \text{drop} \end{array} \right] \pm \left[\begin{array}{c} \text{body force} \\ \text{head} \end{array} \right] \quad (3-41)$$





CONCEPT 1 - COMPOSITE SCREEN WICK



CONCEPT 2 - I-BEAM WICK

S-77563

Figure 3-41. Candidate Wick Designs



The heat pipe can operate without drying out the wick as long as the capillary pumping pressure term is greater than or equal to the pressure losses in the liquid or vapor, and those due to gravitational or acceleration field effects (which may help, hinder, or not exist). Each of these pressure terms is considered individually in the following paragraphs.

The capillary head for a square mesh screen has been shown to be independent of the contact angle, and, hence is predicted by the equation

$$\Delta P_c = \frac{2\sigma}{r_c} \quad (3-42)$$

where σ is the surface tension and r_c is the effective pore radius. For single layers of screen, most of the available data were shown to be successfully correlated by the following equation for the effective pore radius (e.g., Ref. 3-2).

$$r_c = \frac{d + \delta}{2} \quad (3-43)$$

where d is the wire diameter and δ is the wire spacing.

Since in the present concept (composite screen wick) the wire spacing of the fine mesh screen is much smaller than the coarse mesh screen it is treated as a single layer.

The liquid pressure drop is predicted by the following correlation

$$\Delta P_\ell = \frac{Q \mu_\ell (L + L_a)}{2 \rho_\ell \bar{K} h_{fg} a_c g_c} \quad (3-44)$$

Where

Q = heat flux

μ_ℓ = liquid viscosity

ρ_ℓ = liquid density

h_{fg} = latent heat of vaporization

L = heat pipe length

L_a = adiabatic length of heat pipe

$\bar{K}$ = permeability of the screen matrix

g_c = gravitational constant

and a_c = total cross sectional area of the wick



The vapor pressure drop is predicted by, Reference 3-3,

$$\Delta P_v = \frac{4\pi Q \mu_v (L+L_a)}{\rho_v h_{fg} a_v^2 g_c} \quad (3-45)$$

where the subscript v in this case refers to vapor.

Finally, the body force head is given by, Reference 3-3,

$$\Delta P_g = \rho_l \frac{g}{g_c} [D_i \sin \theta + L \cos \theta] \quad (3-46)$$

where θ is the angle the heat pipe makes with the vertical and D_i is the inner diameter of the heat pipe.

In the present application, the ammonia vapor pressure is quite high at the operating temperature which results in high vapor density, and, consequently, low vapor pressure drop as compared to the liquid pressure drop. Hence, the vapor pressure drop ΔP_v can be neglected. For horizontal operation the second term in Equation 3-46 can be neglected and the body force head reduces to

$$\Delta P_g = \rho_l \frac{g}{g_c} D_i \quad (3-47)$$

Hence, the pressure drop criterion becomes

$$\frac{2\sigma}{\left(\frac{d+\delta}{2}\right)} - \rho_l \frac{g}{g_c} D_i = \frac{Q_{\max} \mu_l (L+L_a)}{2 \rho_l \bar{K} h_{fg} a_c g_c} \quad (3-48)$$

The heat transport capability of the heat pipe can be obtained from Equation 3-48 as follows:

$$Q_{\max} = \left(\frac{\rho_l \sigma h_{fg} g_c}{\mu_l} \right) \cdot \left(\frac{\bar{K} a_c}{L+L_a} \right) \left(\frac{8}{d+\delta} - \frac{2\rho_l D_i g}{\sigma g_c} \right) \quad (3-49)$$

All the parameters on the right hand side of Equation 3-49 are known once the heat pipe diameter is selected. The value of the permeability $\bar{K}$ was determined on the basis of experimental data obtained on the ambient heat pipe for the 5 watt VM refrigerator. These $\bar{K}$ values compare quite well with those published for similar wick designs.

Equation 3-49 was used to predict the maximum power transport capability for various heat pipe diameters and lengths. Two heat pipe diameters were selected for this parametric study (0.0127 (0.5) and (0.00952 m (0.375-in.)) covering the range of interest. The upper bound of the range is determined by



the sump length allocated for the heat pipe assembly (0.0556 m (2.1875-in.)), the desired redundancy which requires a multi-heat pipe approach, and the sump diameter which determines the radius of curvature of the heat pipe. The lower bound of the diameter range is dictated by the simplicity of fabrication and assembly.

The length of the ambient heat pipe controls the location of the VM refrigerator in relation to the space radiator on a particular spacecraft. Hence, the effect of the heat pipe length on its maximum power transport capability was investigated.

The length of the condenser end of the heat pipe, and hence the radiator length was assumed constant at 0.61 m (2-ft). The adiabatic length of the heat pipe was changed to vary the overall length of the heat pipe.

The results of this analysis are shown in Figure 3-42 for a heat pipe in the horizontal position. The operating temperature of the heat pipe was assumed to be 332°K (120°F).

5. Temperature Drop

The temperature drop

$$\Delta T = \Delta T_{e,w} + \Delta T_e + \Delta T_v + \Delta T_c + \Delta T_{c,w} \quad (3-50)$$

where $\Delta T_{e,w}$ = temperature drop in the heat pipe wall at the evaporator end

ΔT_e = temperature drop in the wick-liquid matrix (if adjacent to wall) at the evaporator end

ΔT_v = temperature drop in the vapor

ΔT_c = temperature drop in the wick-liquid matrix (if adjacent to wall) at the condenser end

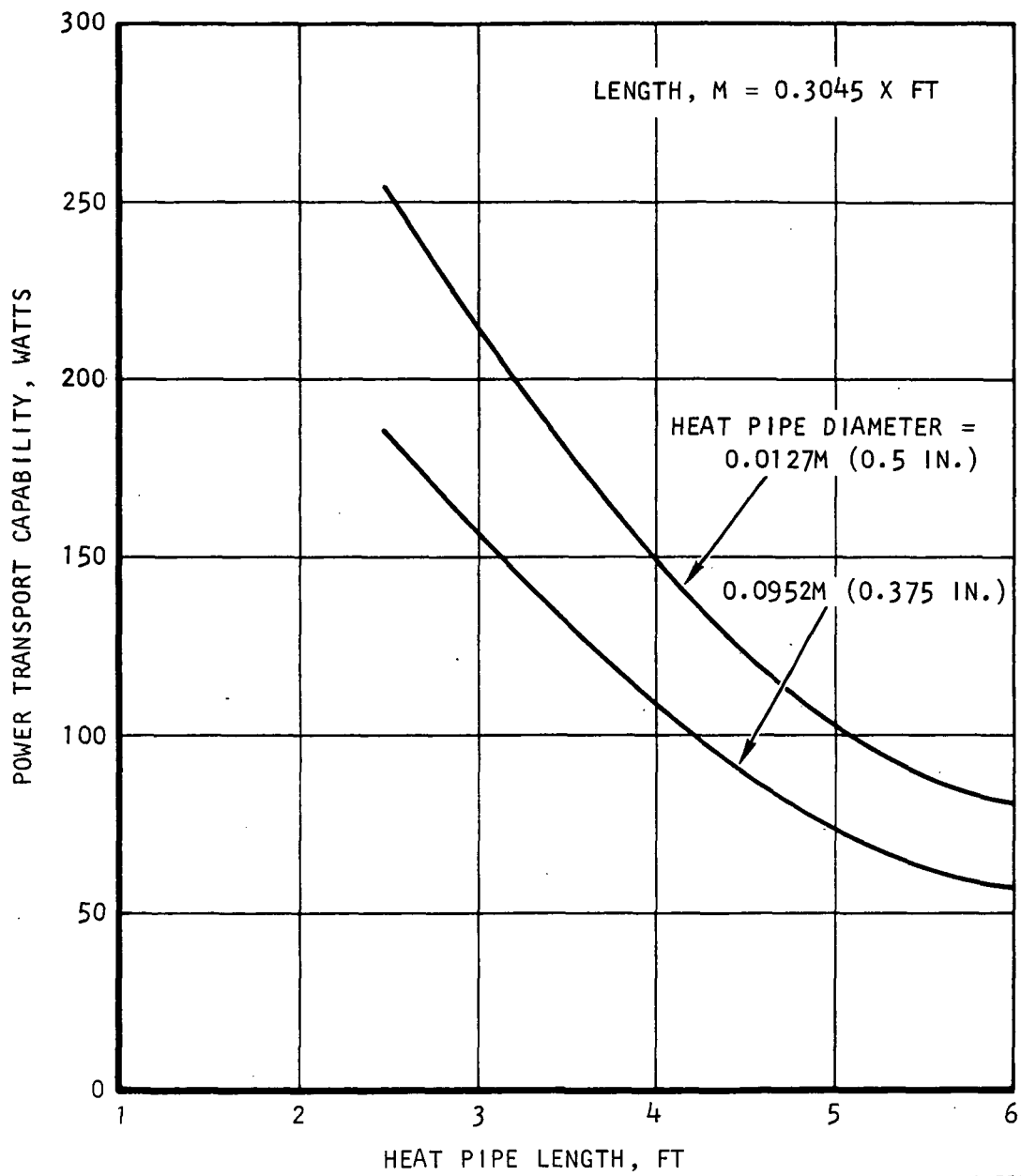
$\Delta T_{c,w}$ = temperature drop in the heat pipe wall at the condenser end

The vapor ΔT_v is usually quite small and can be neglected.

For a round heat pipe, with a wick design similar to that shown in Figure 3-41 Concept 1, the ΔT can be predicted by the following equation

$$\Delta T = Q \left[\frac{1}{\pi k_w \ell_e} \ln \frac{r_o}{r_i} + \frac{1}{\pi k \ell_e} \ln \frac{r_i}{r_{ii}} + \frac{1}{2\pi k \ell_c} \ln \frac{r_i}{r_{ii}} + \frac{1}{2\pi k_w \ell_c} \ln \frac{r_o}{r_i} \right] \quad (3-51)$$





S-77539

Figure 3-42. Effect of Heat Pipe Length on Power Transport Capability



where r_o = heat pipe outside radius
 r_i = heat pipe inside radius
 r_{ii} = wick inside radius
 l_e = evaporator length
 l_c = condenser length
 k_w = wall thermal conductivity
 K = effective thermal conductivity of the wick-liquid matrix
 Q = heat rate transported by the heat pipe

In obtaining Equation 3-51, the following assumptions have been made

- (a) Heat input to the heat pipe at the evaporator end is over half the circumference only.
- (b) Heat rejection from the heat pipe at the condenser end is over the entire circumference.
- (c) The transfer process in the wick is molecular conduction, which is substantiated by test data on an identical heat pipe design.
- (d) Temperature drops in the vapor stream, and at the vapor-liquid interfaces in both the evaporator and condenser, are negligible.

All parameters in Equation 3-51 are usually known a priori except K . The value of K for the present application was determined on basis of the data obtained on the identical design used on the 5-w refrigerator. A sample of this data is shown in Figures 3-43 and 3-44. A schematic of the test system used to obtain this data is shown in Figure 3-45.

The heat pipe ΔT for the present design is shown in Figure 3-46 versus the power transported for two heat pipe diameters 0.00952 (0.375) and 0.0127 m (0.5-in.) and subject to the previous assumptions. The increase in ΔT with the decrease in heat pipe diameter is due mainly to the decrease in the thermal resistance at the evaporator end.

6. Selection of Size and Number of Heat Pipes

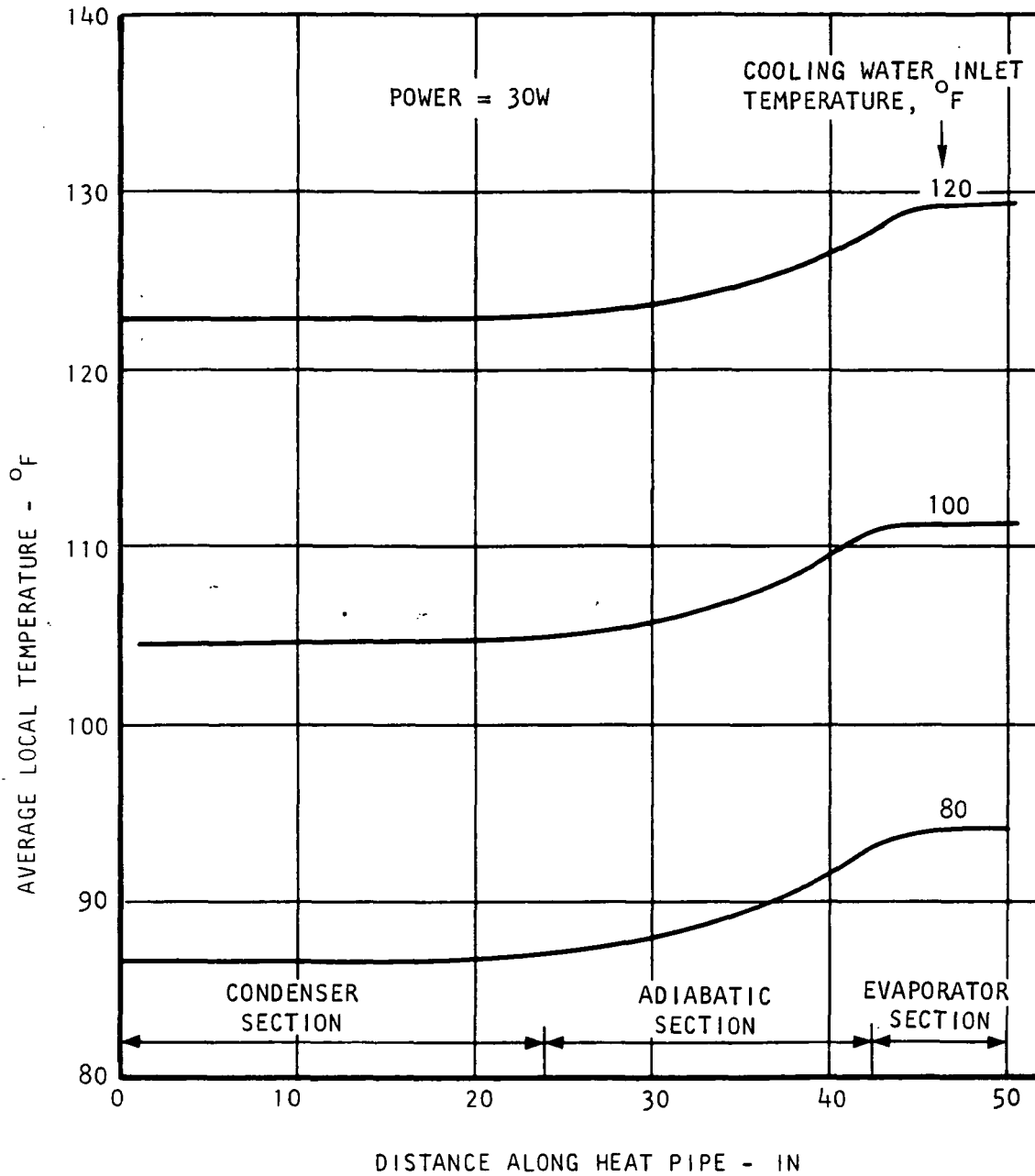
The 50-percent redundancy requirement necessitates the use of several individual heat pipes in the ambient heat pipe assembly. The assembly must be designed such that if 50-percent of the heat pipes fail, the heat pipe assembly will continue to transport a total of 70-w to the radiator while maintaining the sump skin temperature at a maximum value of 334°K (140°F).



NOTES:

1. TEMP, $^{\circ}\text{K} = (^{\circ}\text{F} + 460) / 1.8$

2. DISTANCE, $\text{m} = 0.0254 \times \text{IN.}$



S-77550

Figure 3-43. Effect of Temperature Level on Heat Pipe Performance



- NOTES: 1. $\text{TEMP } ^\circ\text{K} = (^{\circ}\text{F} + 460)/1.8$
 2. $\text{DISTANCE, m} = 0.0254 \times \text{IN.}$

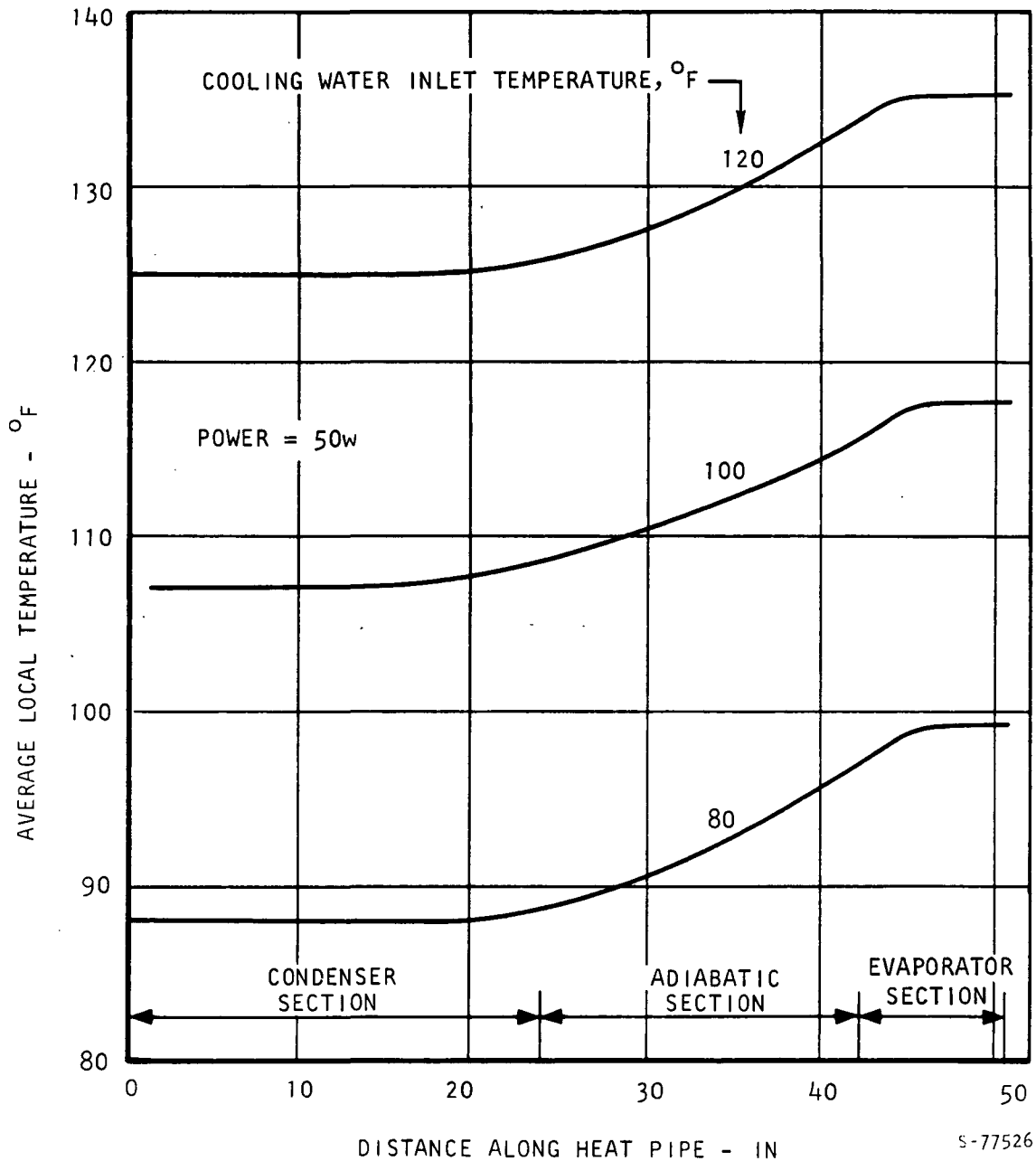
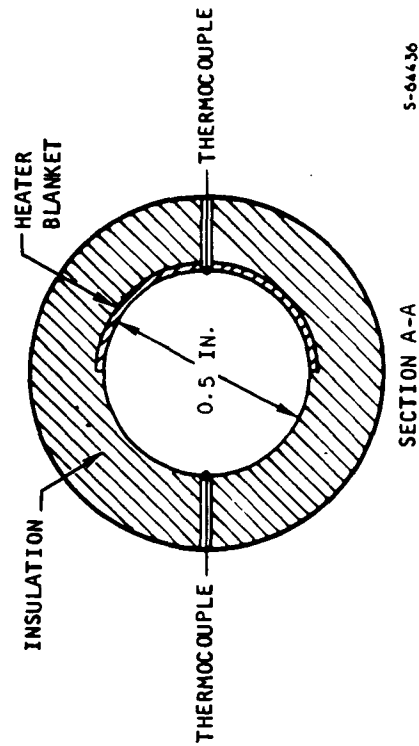
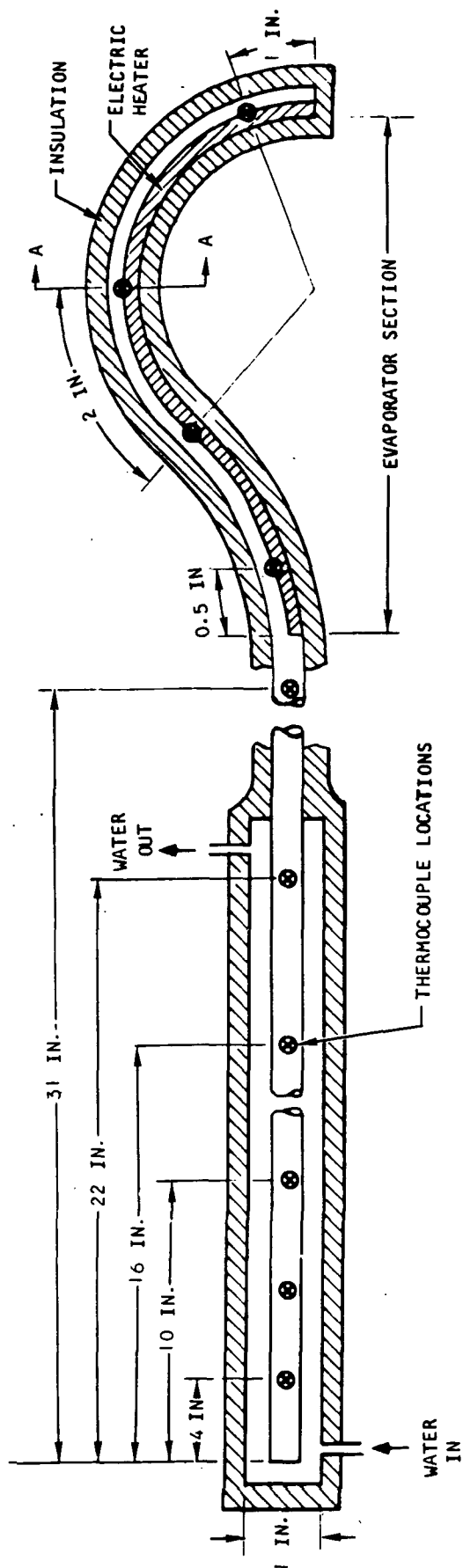


Figure 3-44. Effect of Temperature Level on Heat Pipe Performance

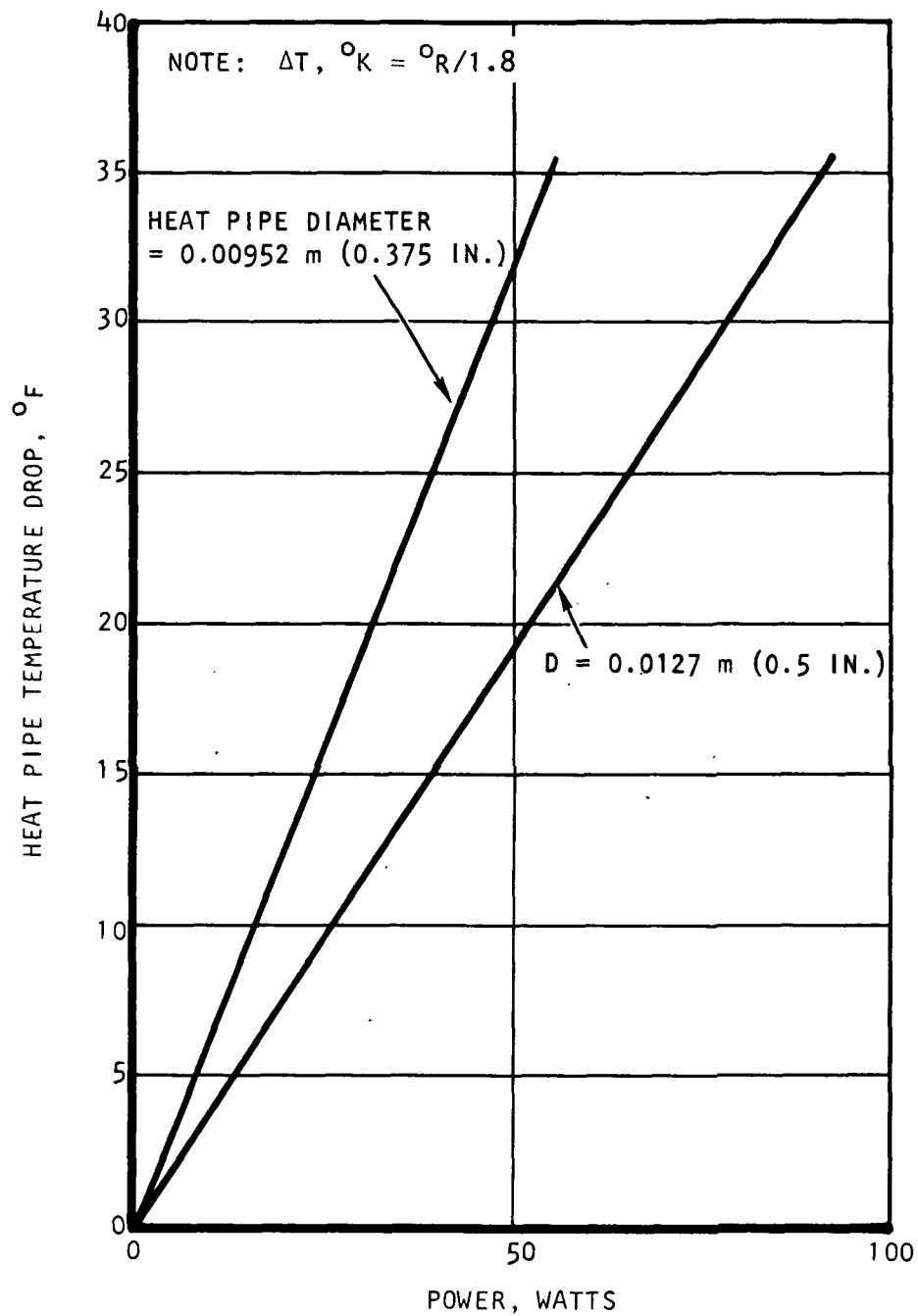




S-64436

NOTE: LENGTH, $m = 0.0254 \times \text{IN.}$

Figure 3-45. Schematic of the Test System



S-77582

Figure 3-46. Overall Temperature Drop vs Power Transported



Figure 3-42 indicates that an assembly using two heat pipes would be able to meet the power transport and redundancy requirements up to a heat pipe length of about 1.525 m (5-ft). On the other hand, Figure 3-46 indicates that if two heat pipes are used and one fails the resulting heat pipe ΔT would range from 15 (27) to 23.9°K (43°F). Adding to this value the temperature drops at the various interfaces the overall temperature drop (from sump to radiator) would significantly exceed the maximum value (16.68°K (30°F)) selected on the basis of the sink temperature study.

Hence, a four heat pipe assembly appears to be optimum. In case of a 50-percent failure each heat pipe would transport 35-w and the heat pipe ΔT would range from 7.22 (13) to 11.67°K (21°F), depending on the diameter selected.

Finally, a heat pipe diameter of 0.00952 m (0.375-in.) would result in a lighter heat pipe assembly and at the same time can meet the power transport and temperature drop requirements. Furthermore the small radius of curvature of the heat pipe at the sump (0.0318 m (1.25-in.)) favors the smaller heat pipe diameter.

Hence, a heat pipe assembly consisting of four aluminum heat pipes of 0.00952 m (0.375-in.) O.D. and using anhydrous ammonia as a working fluid has been selected.

Heat Pipe Assembly Design

The four heat pipes are mounted in an aluminum block which is made in two halves, each containing two heat pipes. The basic design approach of the aluminum block is similar to the copper assembly employed on the 5-w refrigerator. Aluminum was selected for its weight advantage. At the other end of the heat pipe assembly the individual heat pipes interface with the space radiator.

1. Aluminum Block Design

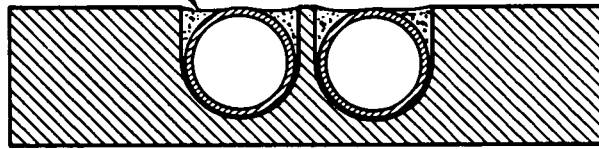
The aluminum block design, depicted in Figure 3-47 employs an interface clamp design which is adaptable to use with either a water cooling system or an ambient heat pipe assembly. Either the cooling coil or the individual heat pipes are soft soldered to the aluminum block, as shown in Figure 3-47, thus allowing the interchange of either system. The soft solder provides good thermal contact between the heat pipes and the aluminum and, also, allows heat to be distributed over most of the pipe circumference.

The two halves are mounted on the engine sump with indium foil (0.0000762 m (0.003-in.) thickness) at the interface and four bolts located in slots as shown in Figure 3-47. This scheme provides good thermal contact between the block and the sump while providing detachability of either the heat pipe assembly or the cooling water assembly.

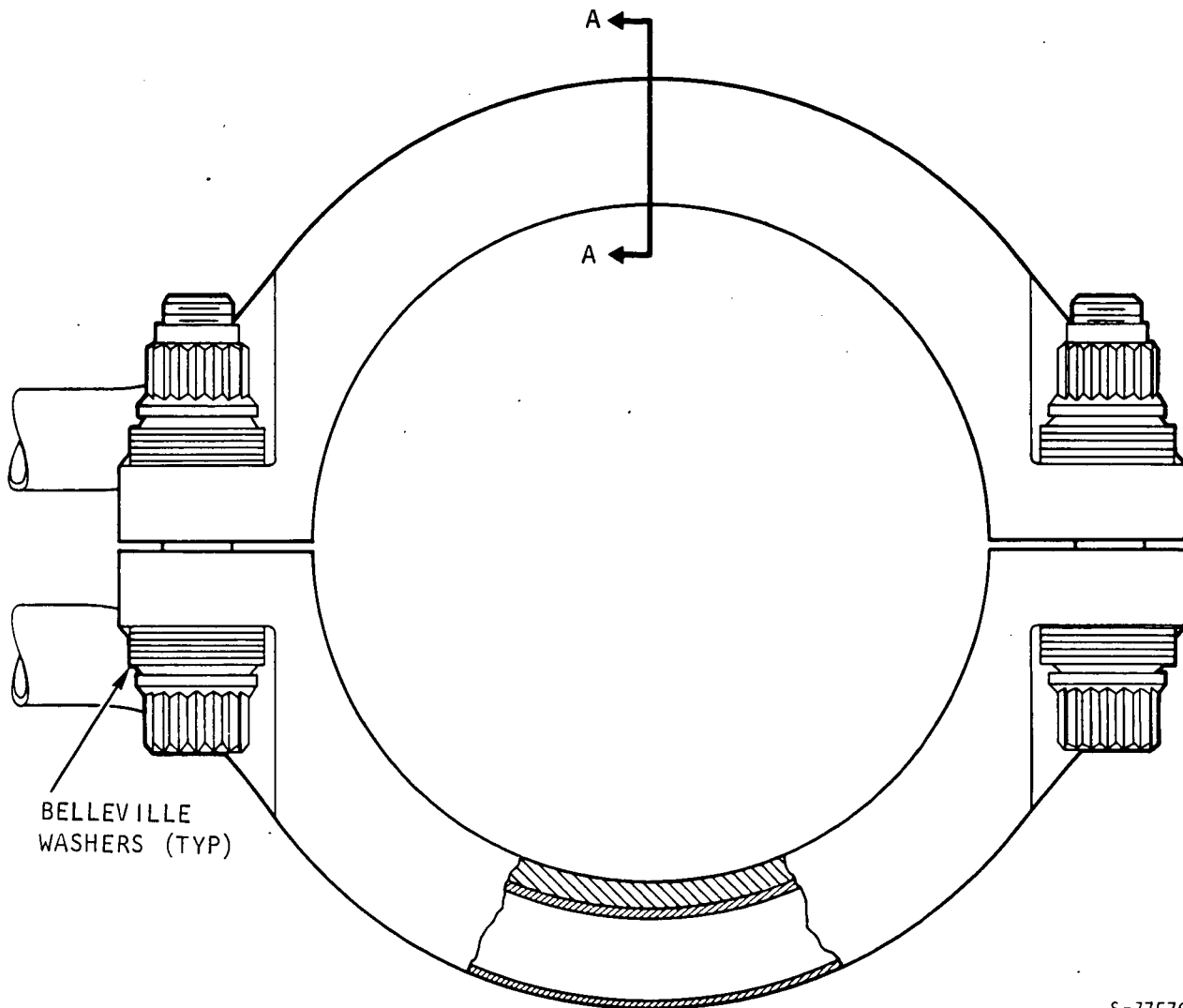
The bolts use compression washers under each bolt head and each nut to achieve an interface pressure in the range of 3.477×10^5 (50) to 6.895×10^5 N/m² (100 psi) (between the engine sump and aluminum block). The fluctuations



HEAT PIPE SOFT SOLDERED
TO ALUMINUM BLOCK



SECTION A-A



BELLEVILLE
WASHERS (TYP)

S-77570

Figure 3-47. Aluminum Block Design Details



AIRESEARCH MANUFACTURING COMPANY
Torrance, California

in the engine sump pressure will result in an increase or decrease of the deflection experienced by the washers while maintaining the interface pressure in the desired range.

Interface Temperature Drop

1. Heat Pipe Operation

The temperature drop at the various interfaces and in the aluminum block are added to the heat pipe temperature drop to give the overall temperature drop in the ambient heat pipe assembly, $\Delta T_{\text{overall}}$. Hence,

$$\Delta T_{\text{overall}} = \Delta T_{i,1} + \Delta T_{A1} + \Delta T_{i,2} + \Delta T + \Delta T_{i,3} \quad (3-52)$$

where $\Delta T_{i,1}$ = temperature drop at the interface between the sump and the aluminum block

ΔT_{A1} = temperature drop in the aluminum block

$\Delta T_{i,2}$ = temperature drop at the interface between the aluminum block and the heat pipe

ΔT = temperature drop in the heat pipe

$\Delta T_{i,3}$ = temperature drop at the interface between the heat pipe and a space radiator

The thermal resistance at an indium foil interface was based on the data in Reference 3-4. For interface pressure between 3.477×10^5 (50) and 6.895×10^5 N/m² (100 psi) the interface thermal resistance has the value

$$R_{i,1} = 4.89 \times 10^{-6} \frac{^{\circ}\text{K} \cdot \text{m}^2}{\text{W}} \left(0.004 \frac{^{\circ}\text{F} \cdot \text{in.}^2}{\text{Btu/hr}} \right)$$

which gives

$$\Delta T_{i,1} = 0.0556^{\circ}\text{K} \quad (0.1^{\circ}\text{F})$$

The soft solder joint thermal resistance was obtained from the data by Yovanovich and Tuarze, Reference 3-5. Hence,

$$R_{i,2} = R_{i,3} = 1.505 \times 10^{-5} \frac{^{\circ}\text{K} \cdot \text{m}^2}{\text{W}} \left(0.042 \frac{^{\circ}\text{F} \cdot \text{in.}^2}{\text{W}} \right)$$



Assuming two heat pipes to be carrying the entire 70 w results in

$$\Delta T_{i,2} \cong 0.334^{\circ}\text{K} (0.6^{\circ}\text{F})$$

and $\Delta T_{i,3} = 0.278^{\circ}\text{K} (0.5^{\circ}\text{F})$

The temperature drop at the radiator interface assumes a condenser length of 0.61 m (2-ft) and that the radiator panel interfaces with one side of the heat pipes over an arc length of 0.000254 m (0.1-in.).

Finally the aluminum block temperature drop, under failure of 2 heat pipes, is estimated to be

$$\Delta T_{A1} = 5.36^{\circ}\text{K} (10^{\circ}\text{F})$$

Hence, the overall ΔT in the ambient heat pipe assembly is estimated to have a maximum value of

$$\Delta T_{\text{overall}} \cong 16.4^{\circ}\text{K} (29.5^{\circ}\text{F})$$

2. Water Cooled Operation

The design of the aluminum block is such that water cooling coils may be substituted directly for the four heat pipes. The water cooling coils will be of the same diameter and geometric configuration as the evaporator end of the heat pipes.

An analysis of the water cooling coils was performed similar to that for the 5 watt VM. The analysis indicated that the cooling water must pass through the four sections of the water cooling coil in series. The water flow was assumed at $0.00315 \text{ m}^3/\text{sec}$ (50 gal/hr). The maximum temperature drop from the sump gas to the cooling water with this configuration is approximately 6.67°K (12°R), which is less than the temperature drop during heat pipe operation.

MECHANICAL DESIGN

The mechanical design of a long life Vuilleumier refrigerator requires detailed attention to every part in order to assure meeting the design requirements. This section presents a description of the various parts of the machine, and detailed discussions of the interfacing requirements with the Honeywell radiometer and detector assembly. The preliminary design stress analysis is also discussed, as are the machine closure techniques.

Design Description

The preliminary design generated during Task 1 has drawn heavily on the experience gained during the design and development of a similar device for



GSFC. Under a former contract to GSFC, an engineering prototype of a 5-watt Vuilleumier Cryogenic Refrigerator was developed that met requirements very similar to those required under this current contract. In order to enhance the reliability and performance characteristics of the fractional watt flight prototype refrigerator, similar or identical design approaches to the 5 watt machine have been employed.

Description of the functional characteristics may best be accomplished by treating each detail of the design independently. The basic geometry, function, and materials of construction of each detail are described; these components are also related to the overall device.

1. Hot-End Insulation System

Thermal containment of the heat supplied to the hot end is accomplished by applying Johns Manville Min-K insulation spirally wrapped around the hot-cylinder followed by a thin layer of AA-1200 fiberglass to fill out the blanket-ing to the outer protective shell. This shell will be fabricated of a corrosion resistant alloy. The cylinder will be terminated at one end in an ellipsoidal dome and at the opposite end in an O-ring-weld flange. This protective vessel will be attached to an evacuation system, thereby allowing the reduction of the pressure within the insulation system to a level of 10^{-4} torr or less to provide optimum thermal performance.

2. Heater-Heat Exchanger Complex

The introduction of heat into the VM hot end will be accomplished by a Cal-rod type of heating element. These devices are inherently long lived and reliable when properly installed. The heater is to be brazed to the elliptical dome transition from the cylindrical regenerator to the hot displacer hot bearing housing. Inside this elliptical surface a channel type forced convection heat exchanger has been provided to direct the flow into and out of the hot displaced volume of the refrigerator. The Cal-rod unit is composed of a corrosion resistant outer sheath with the heating element contained in a swaged magnesium-oxide insulator. The pressure wall dome is an integral part of the hot end pressure vessel and is fabricated of heat treated Inconel 718. The heat exchanger liner also will be fabricated from heat treated Inconel 718.

3. Hot Displacer Hot Bearing System

The hot displacer is supported by bearings located at each end. To provide the long life sliding bearing required at the hot end of this refrigerator, Boeing Compact 6-84-1 is to be utilized working against a chrome carbide-nickel chrome matrix flame sprayed surface. A similar system was utilized in the 5-watt VM refrigerator. The Boeing Compact is assembled into a journal bearing with a metallic sheath for protection. This is then captured in the Inconel 718 pressure vessel housing and maintained in axial position by a high rate spring and retainer system. The chrome carbide mating surface is provided by flame spraying directly onto the journal provided by the displacer. This is then finish ground to the appropriate dimension.



4. Hot Regenerator

The hot regenerator is formed by the installation of precise sheared 100 mesh stainless steel screen discs. These are bounded on the outside by the heat treated Inconel 718 pressure vessel wall and on the inside by the regenerator inner liner. The liner is formed by extending the hot end heat exchanger into the cylindrical section thereby forming the inner diameter junction for the regenerator. This element simultaneously forms the outer wall for the displacer bore and the reference cylinder for the displacer labyrinth seal. The cool end of the hot regenerator is terminated by a flow distribution plate which uniformly directs the flow into and out of the regenerator from the sump heat exchanger section. This distributor plate also retains the screen disks both during assembly and operation of the regenerator.

5. Hot Displacer

This component is to be fabricated from heat treated Inconel 718 to provide the strength required and also to assure a compatible coefficient of expansion with that of the regenerator inner liner. The displacer is supported by a bearing system at each end. The hot end was described in the hot bearing discussion. The sump end will be similarly supported by a Boeing Compact 6-84-1 sliding journal bearing mating with a tungsten carbide surface. This surface is to be flame sprayed onto the Inconel 718 displacer base material, similar to the technique utilized on the hot end, then finish ground to size.

The displacer itself is composed of two pieces; welded at the central support rib. This allows machining of the two pieces to provide a minimum weight, maximum strength system. Convective heat transfer within the displacer is reduced by the inclusion of fibrous quartz prior to the closure weld. Labyrinth seals are to be integrally machined at the hot end of the cylindrical section to affect a pneumatic seal for the helium working fluid.

6. Sump Section

The region from the hot regenerator to the cold regenerator has been designated the sump region. One of the major components contained in this region is the ambient heat exchanger. This is formed by brazing an offset copper fin to the ID of the Inconel 718 pressure vessel cylindrical section. The inner wall of this heat exchanger is formed by the filler block installed in the sump area to reduce the dead volume. The filler block is to be fabricated from high strength aluminum alloy, and will enclose and support the cold end bearings for the hot regenerator. Flow passages are to be provided in the filler block leading from the cavity behind the hot displacer to the sump heat exchanger. The crankshaft area is also ported to the heat exchanger.

7. Crankshaft

High strength heat treatable alloy will be used to fabricate the multi-piece crankshaft. The concept developed for use in the fractional watt VM is quite similar to that utilized in the 5 watt refrigerator fabricated under



another GSFC contract. For this application, individual cam members and main bearing journals are to be pinned together for positive orientation and centrally retained by a high strength locking screw. This technique was adopted in order to allow installation of the crankshaft--connecting rod complex from a central access port. Support of the crankshaft is to be accomplished utilizing Boeing compact 6-84-1 sleeves running against tungsten carbide flame sprayed on the crankshaft journals and ground to size. Similarly, the connecting rod journals will be flame sprayed with tungsten carbide and ground to size. The connecting rod bearing mating surfaces will also be composed of solid Boeing Compact 6-84-1 journals pressed into the connecting rod ends.

Axial positioning of the crankshaft for alignment is to be accomplished by a system of shimming and finally retained in the proper position by a high rate spring system. This latter device is to be incorporated to facilitate the installation and removal of the auxiliary break in magnetic drive without disturbing the critical location of the crankshaft.

8. Connecting Rod-Wristpins

The concept of two piece split connecting rods will also be adopted from the 5 watt design. This is to allow final assembly from a central access port. Material is to be titanium in order to provide the required strength and also light weight for reduced rotating mass. The wrist pins are to be made of solid tungsten carbide and these will be running against Boeing Compact 6-84-1 journals which are an integral part of the connecting rod end.

9. Cold Displacer

The construction of the cold displacer is similar to that of the hot. The displacer body is to be fabricated from four sections, welded and finish ground to size. The displacer utilizes the cantilever concept of support to eliminate the need for the use of low temperature bearings. The bearing surface is to be flame sprayed tungsten carbide, ground to size, running against Boeing Compact 6-84-1. As with the hot displacer, the material is to be Inconel 718 in order to provide the expansion compatibility necessary in a device which must operate through high thermal excursions with small operating clearances. The cold end labyrinth seal is integrally machined into the warm end of the displacer. The interior of the displacer is packed with fibrous quartz.

10. Cold Regenerator

The use of monel shot 0.000178 and 0.000254 m (0.007 and 0.010 inches) in diameter will provide the maximum regeneration with a minimum of axial conduction. Each end of the regenerator will employ 150 mesh monel screens to retain the ball packings and to provide radial flow distribution. The inner wall of the regenerator is formed by a thin wall Inconel 718 tube which also acts as the stationary portion of the labyrinth seal.

A flow distribution annulus will be provided at the cold end of the regenerator and a radial slot distributor is to be utilized at the warm end. Both of these devices are provided to assure uniform radial and circumferential distribution of the helium fluid and also to prevent channeling.



The outer wall of this annular regenerator is formed by the pressure vessel wall, which in turn is the inner vessel for the evacuated dewar insulation system. Here also, Inconel 718 will be used to provide the lightweight high strength system necessary for containment of the 1000 psi working pressure of the helium.

11. Cold End Insulation

The use of two concentric rhodium plated radiation shields will assure adequate thermal protection of the low temperature section of the refrigerator. The cylindrical shields will be supported between the cold regenerator outer cylinder and the dewar outer cylinder by fiberglass pads. Provisions have been made to allow evacuation to the required 10^{-5} torr for optimum thermal protection performance.

12. Cold End DCA Interface and Heat Exchanger

The cold end heat exchanger is located at the cold end of the cold regenerator, and is composed of axial flow slots. The refrigeration heat load from the detector chip assembly (DCA) must be imparted to the gas at this location with the minimum temperature drop possible. This will be accomplished by a flat surface interface, heavily copper plated for low thermal resistance, and ground to an exacting surface flatness. The DCA is then to be held in place on this surface by three screws utilizing Belleville washers for constant, uniform loading of the interface. To enhance this interface thermally, thin foil indium will be used to assure uniform, repeatable heat transfer characteristics. The final interface requirement between HRC and AiResearch equipment is that of the lead matrix. This element will be installed by HRC at the time of the DCA integration and will be soldered to the upper flange of the vacuum dewar outer vessel. For temporary evacuation and evaluation at AiResearch, a cold end closure system will be used, which will be described in a later portion of this report.

13. Instrumentation

The instrumentation required to monitor the performance of the VM may be divided into two categories: (1) Development Test and diagnostic instrumentation and (2) flight performance monitoring equipment. In order to provide the tools necessary for initial performance analysis it is anticipated that the VM will be fitted with a complete complement of diagnostic instruments for laboratory evaluation. A listing of the instruments and their function is presented in Table 3-16 and also illustrated in Figure 3-48 to show the relative locations. For a flight configuration refrigerator, eleven of these devices would be retained or installation made compatible with telemetry signal conditioning. These would include:

1. Sump temperature
2. Hot-end temperature
3. Cold end temperature



TABLE 3-16

FRACTIONAL WATT VM DEVELOPMENT TEST INSTRUMENTATION

Designation	Function Measurement	Device	Location
T ₁	Temperature	Thermocouple Cu-Con ↑	Simulated cold end load temp
T ₂	Temperature		Simulated cold end load temp
T ₃	Temperature		Cold end interface flange
T ₄	Temperature		Cold end heat exchanger
T ₅	Temperature		Cold end heat exchanger
T ₆	Temperature		Cold regenerator, cold end
T ₇	Temperature		Cold regenerator, cold end
T ₈	Temperature		Cold regenerator, mid cylinder
T ₉	Temperature		Cold regenerator, mid cylinder
T ₁₀	Temperature		Cold regenerator, warm end
T ₁₁	Temperature	Thermocouple Cu-Con ↓ Thermocouple Ch-Al ↑	Cold regenerator, warm end
T ₁₂	Temperature		Sump heat exchanger, cold end
T ₁₃	Temperature		Sump heat exchanger, cold end
T ₁₄	Temperature		Heat pipe block, cold end
T ₁₅	Temperature		Heat pipe block, mid block
T ₁₆	Temperature		Heat pipe block, warm end
T ₁₇	Temperature		Sump heat exchanger, warm end
T ₁₈	Temperature		Sump heat exchanger, warm end
T ₁₉	Temperature		Hot regenerator, cold end
T ₂₀	Temperature		Hot regenerator, cold end
T ₂₁	Temperature		Hot regenerator, mid cylinder
T ₂₂	Temperature		Hot regenerator, mid cylinder
T ₂₃	Temperature		Hot regenerator, hot end
T ₂₄	Temperature		Hot regenerator, hot end
T ₂₅	Temperature		Hot end heater surface
T ₂₆	Temperature		Hot end heat exchanger
T ₂₇	Temperature		Hot bearing surface
EI ₁	Power	Watt meter	Cold end simulated power in
EI ₂	Power	Watt meter	Drive motor power in
EI ₃	Power	Watt meter	Hot end heater power in
PT ₁	Internal Pressure	Transducer	System helium operating pressure
P ₁	Vacuum pressure	Ion gauge	Cold end insulation vacuum pressure
P ₂	Vacuum pressure	Ion gauge	Hot end insulation vacuum pressure
S ₁	Speed	Magnetic monopole	Engine speed



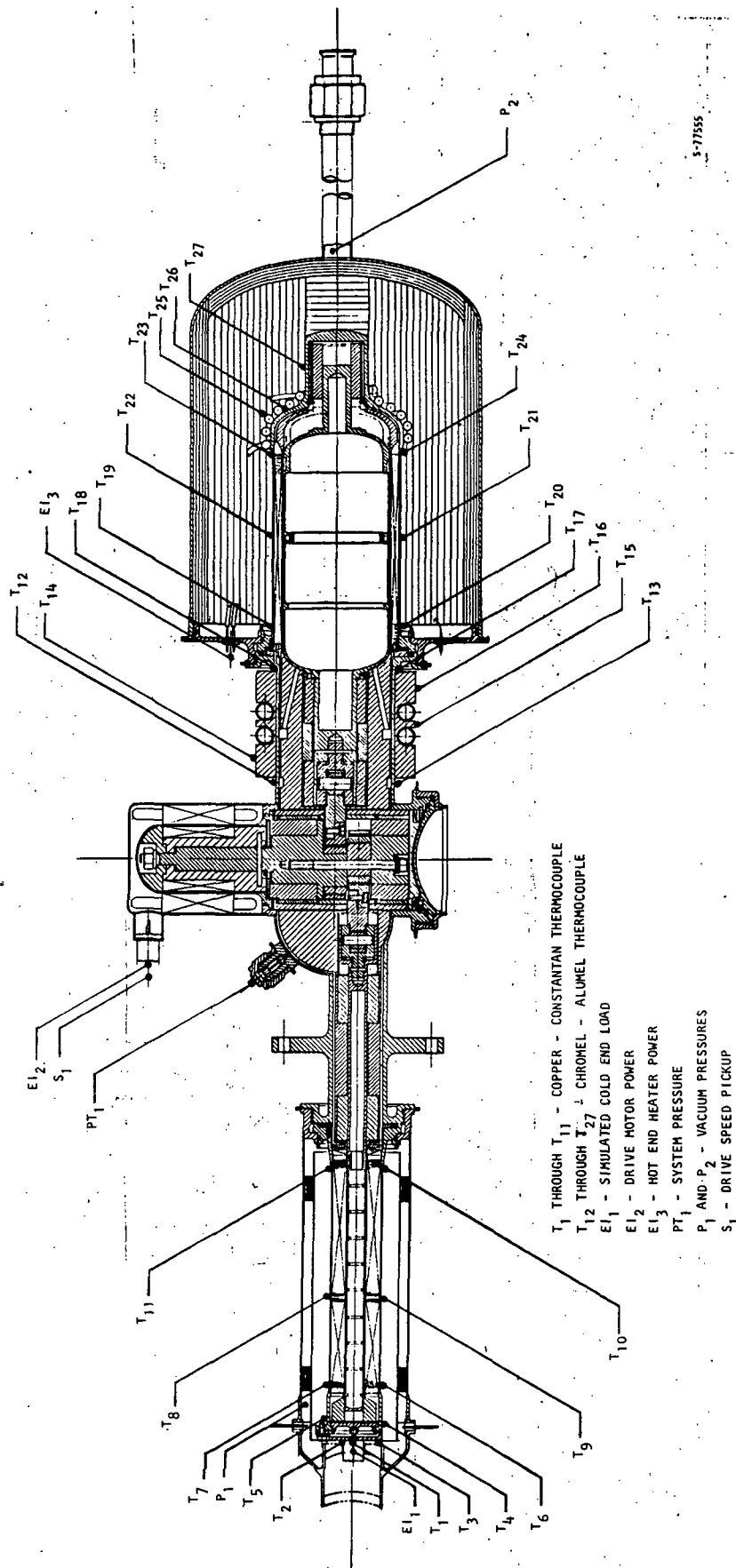


Figure 3-48. Proposed Fractional Watt VM Development
Test Instrumentation

4. Four heat pipe temperatures
5. Internal pressure
6. Motor speed
7. Hot-end power
8. Motor power

14. Auxiliary Bearing Break-In Drive System

During the break-in period of the NASA/GSFC 5 watt VM, it was found that the power required to drive the machine greatly exceeded the original design goal. Following the break-in period, the motor power is expected to return to the design level, based on the power reductions observed during the first 1000 hr of break-in. For this reason, it was deemed necessary to provide a method of breaking in the fractional watt design without overdesigning the airborne motor, which would require excessive power consumption during the entire flight. Therefore, a laboratory break-in motor coupling was designed (Figure 3-49). This design utilizes the magnetic coupling concept originally developed for the 5 watt VM engine, as adapted quite simply to the fractional watt VM. To utilize the break-in drive system, the cover on the outboard crankshaft access port is removed from the fully assembled engine. Removal of this cover in no way affects the position or alignment of any component within the machine since all alignment is maintained by the snap ring located in the bore of the main support tube. Removal of the cover exposes the outboard end of the crankshaft. An auxiliary drive shaft is attached to the crankshaft with three screws. A filler block and a magnetic rotor are mounted on the shaft. A special break-in dome is then placed over the magnetic rotor and the threaded retaining ring installed, thereby resealing the internal volume of the engine. An externally driven magnetic rotor is then installed (quite similar to the design utilized in the 5 watt VM) which in turn is connected to a laboratory drive motor which has approximately ten times the torque capability of the airborne motor.

An additional design feature afforded by this approach is the ability to operate the refrigerator with either the laboratory or the airborne motor. Operation from either motor may be selected without disturbing the test setup or opening the sump section of the VM. When the running torque has reached the design level, the system may be switched from the laboratory to the airborne motor during operation. In the event that the additional inertia of the break-in magnetic coupling precludes starting with the airborne motor alone, startup may be accomplished by utilization of both motors. The laboratory motor will be disengaged when the refrigerator has achieved nominal running speed.

Once engine break-in is complete, the entire magnetic drive mechanism is removed and the normal "flight" bearing access cover installed. In this way, the flight motor may be designed to produce only that torque necessary to drive the engine at the optimized minimum power level.



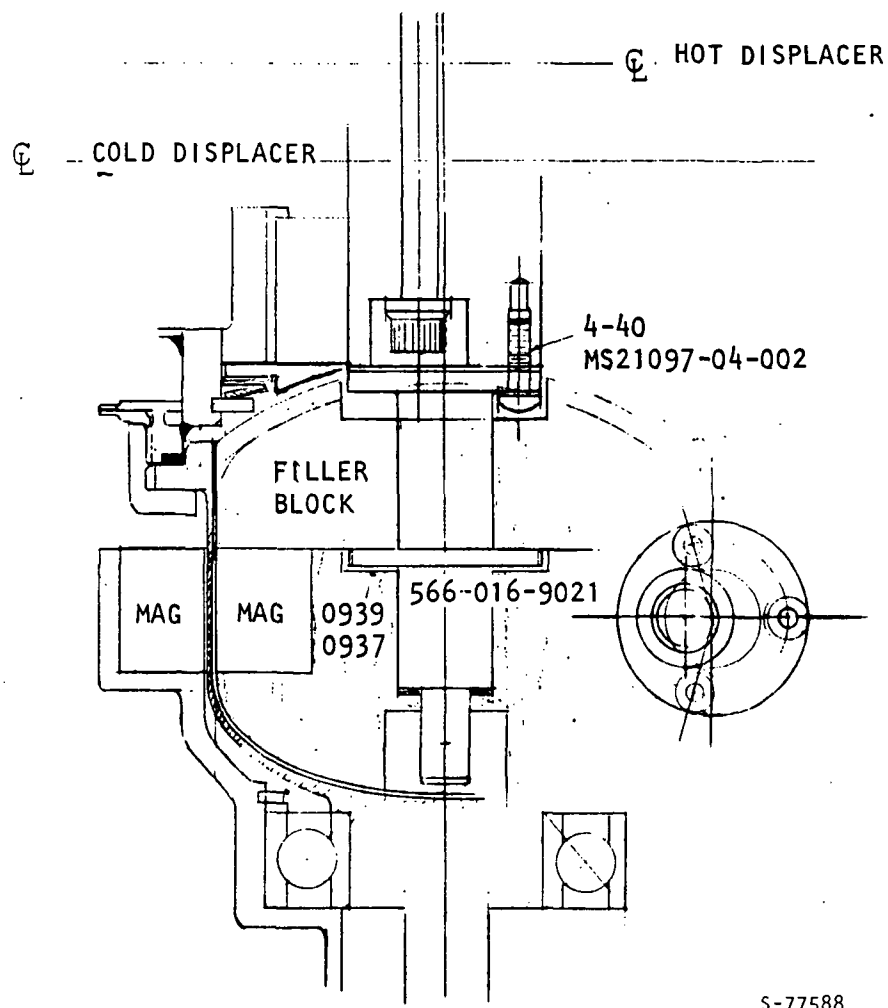


Figure 3-49. Break-In Magnetic Coupling



15. Radiometer and DCA Interfacing Techniques

Several areas require careful design consideration in integrating the VM refrigeration package into the Honeywell radiometer. During the course of the Task I activities, several meetings, telephone conversations and communications were exchanged between AiResearch and Honeywell Radiation Center personnel. The items of information may be grouped into five categories and are discussed below by major topic.

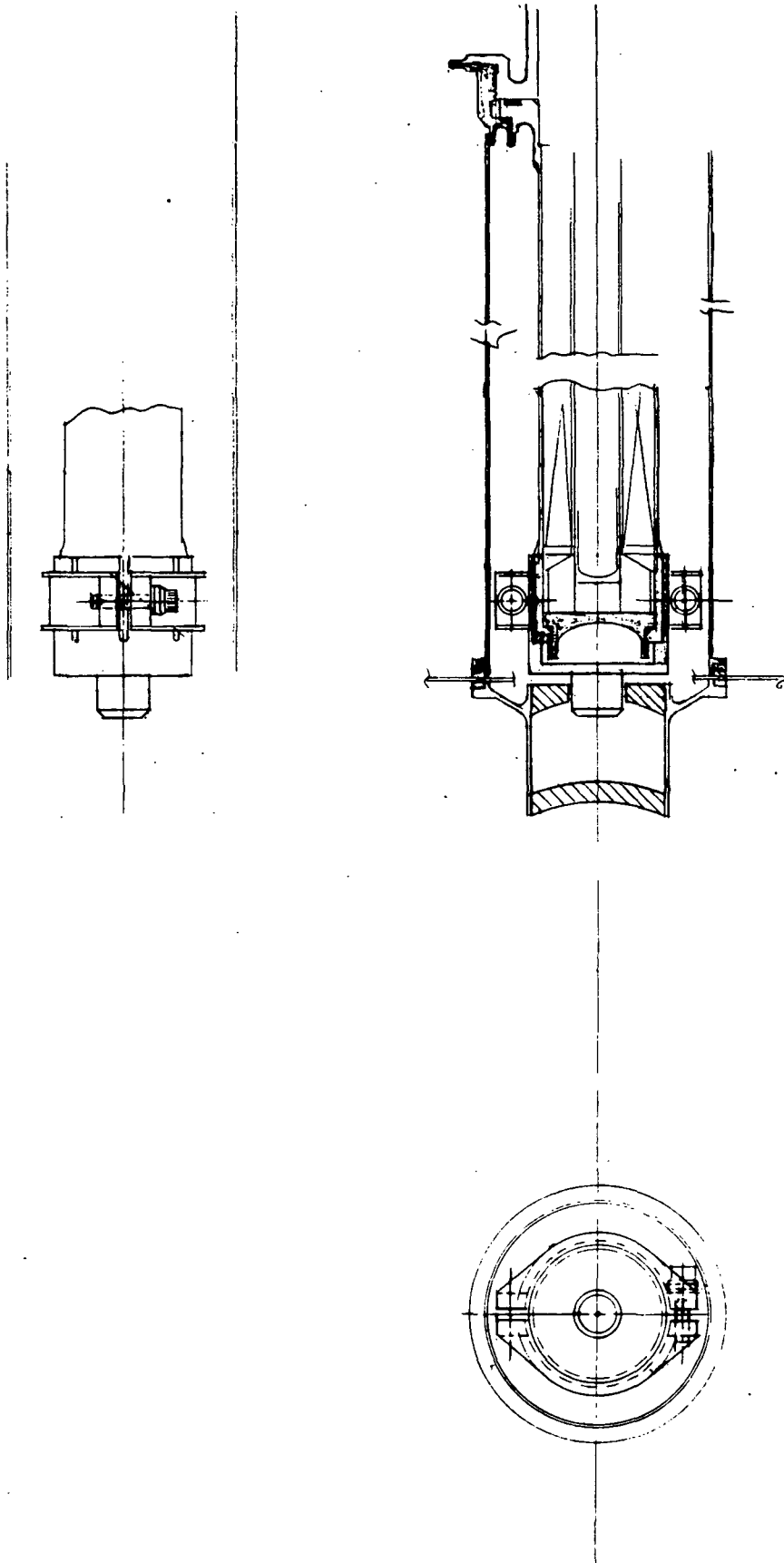
16. VM-DCA Interface

Early discussions regarding the Detector Chip Assembly (DCA) centered around the needs of Honeywell Radiation Center (HRC) to maintain the detector in a precisely located axial and radial position. This leads to the existence of two distinct areas to be considered: (1) axial and radial location statically, and (2) axial and radial location under operating vibratory conditions. This latter facet is to be treated separately and is discussed under the cold end vibration testing of the 5 watt VM refrigerator.

Initial requirements set forth by HRC dictated the axial prepositioning of the DCA within ± 0.0000254 m (± 0.0001 in.). AiResearch studied several interface configurations for the DCA and the VM but it became obvious that the probability of being able to maintain the dimensional location of the DCA interface within the limits required by HRC would be quite remote, due to the fact that a hard interface point (a ledge or shoulder) ground to a very precise dimension, would be required. The DCA would then have to be clamped into place utilizing a heat transfer improvement mechanism such as indium foil. Indium foil in itself caused problems in each of the designs considered. First, the thickness of indium foil varies from batch to batch, as well as within a given sheet, more than the allowable 0.0000508 m (0.0002 in.) required by HRC. Second, the relative surface condition of both the VM and the DCA could influence the degree of indium foil flow in order to accomplish the required thermal interface. Third, the quantity of flow of the indium foil as the result of thermal cycling, coupled with constant mechanical load, would produce a potentially changing position of the DCA.

The results of this AiResearch study were relayed to HRC, with a request that the requirement for an axial interface be reconsidered. It was agreed that no additional effort be expended on the axial concept and that a radial scheme would be incorporated. Based on this agreement, several conceptual layout sketches were generated. Figures 3-50 through 3-54 illustrate several of the approaches considered. Illustrations 3-50 through 3-53 all had one common undesirable feature--namely the indium foil has to be installed on the cylindrical surface and a close fitting copper clamping collar installed over the cold end of the VM and the indium foil. These techniques would, however, provide the axial dimensional prepositioning of ± 0.0000254 m (± 0.0001 in.) required by Honeywell. It was felt that the attractiveness of any of these approaches was considerably decreased by the problem foreseen with the actual installation of the indium foil between the DCA and the VM wherein the foil could be crumpled during installation. Therefore, the method would never provide a positive interface area that could be repeatedly attained.





S-77568

Figure 3-50. DCA-VM Radial Interface - Concept A



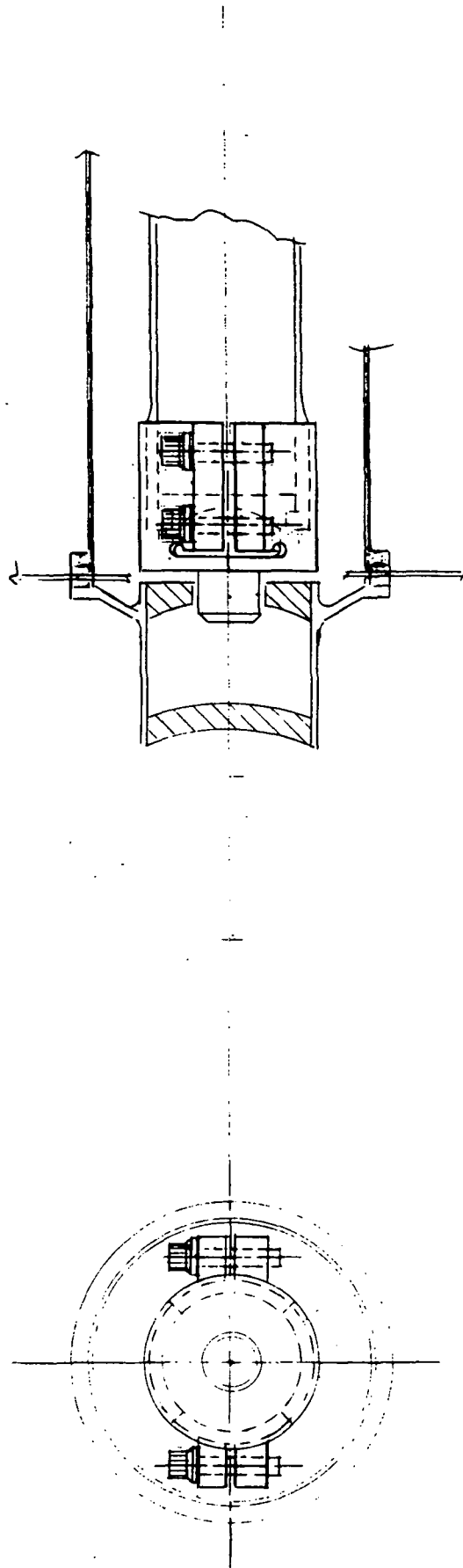


Figure 3-51. DCA-VM Radial Interface - Concept B



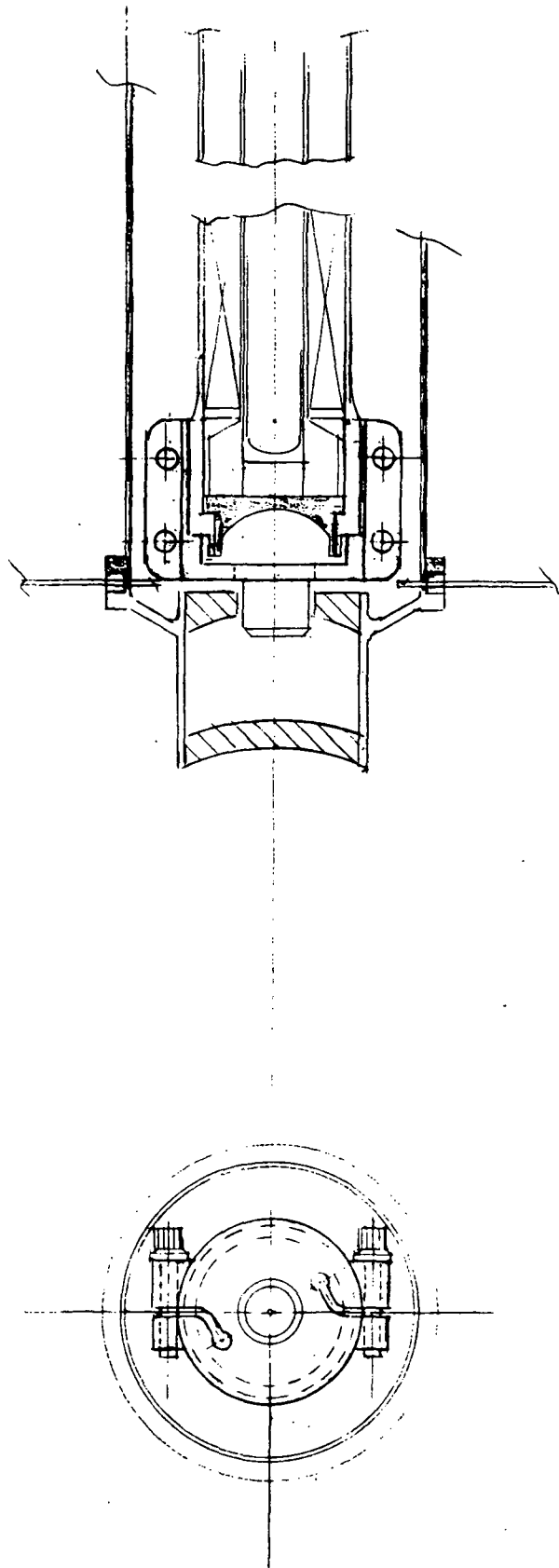


Figure 3-52. DCA-VM Radial Interface - Concept C



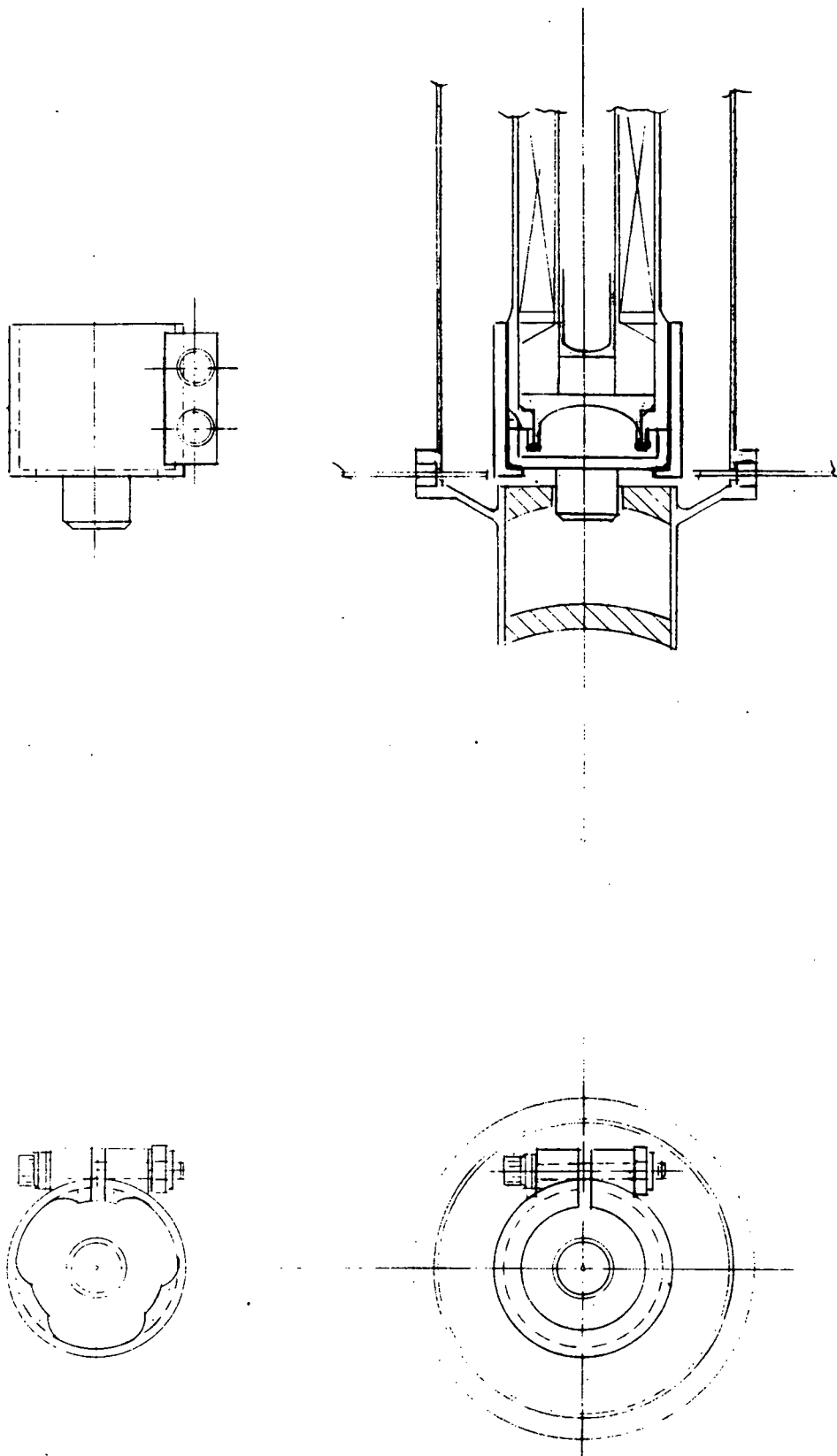
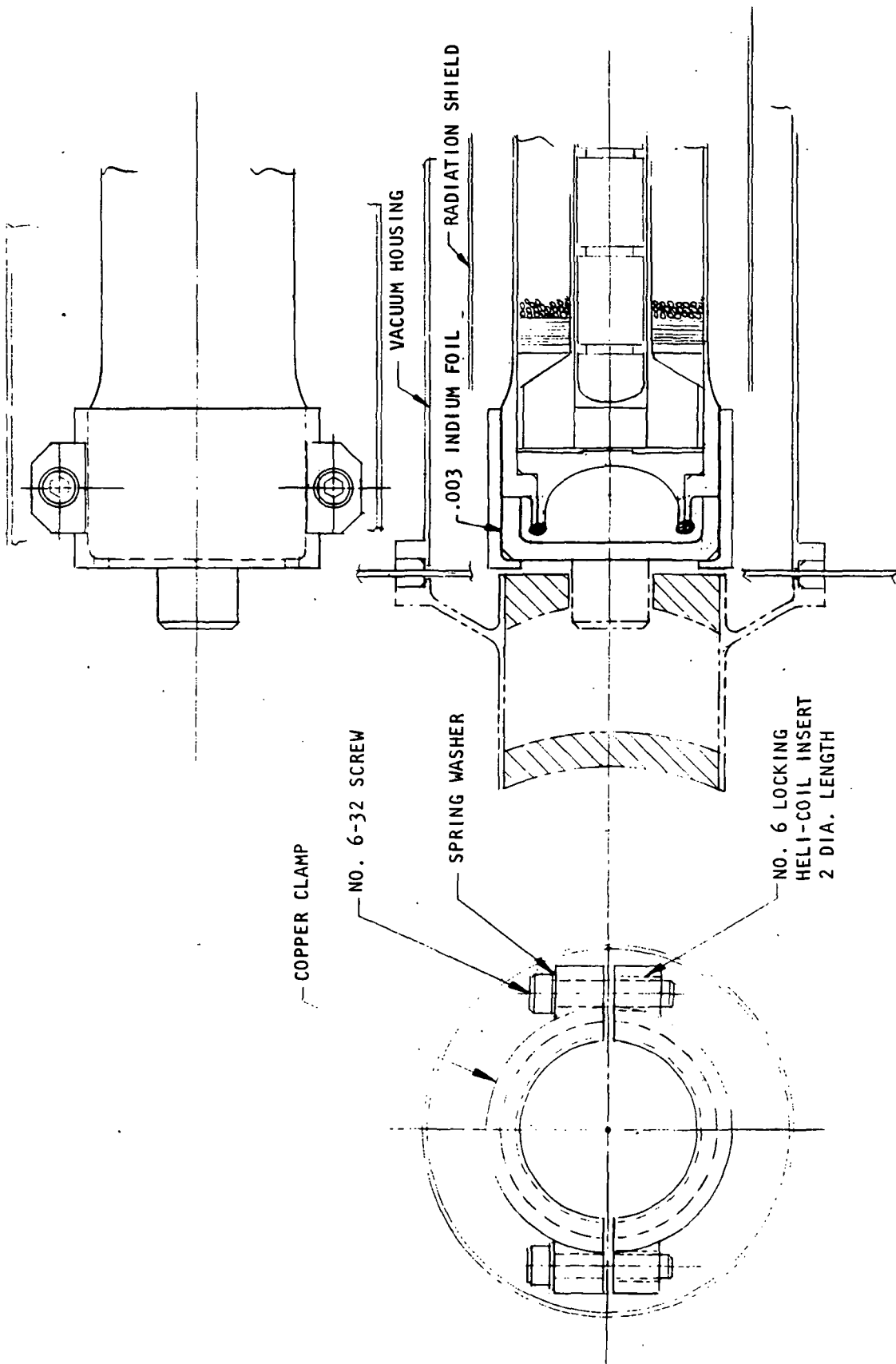


Figure 3-53. DCA-VM Radial Interface - Concept D





s-77566

Figure 3-54. DCA-VM Radial Interface - Concept E



AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California

A method of eliminating this objection is shown in Figure 3-54, but does require precise control of the outside diameter of both the DCA and the VM to assure the uniform interface pressures necessary for adequate heat transfer while simultaneously assuring the axial location of the DCA within the 0.000254 m (± 0.0001 in.) tolerance.

HRC was again contacted and apprised of the problems encountered. It was finally agreed to eliminate the axial tolerance requirements in favor of the axial interface system. Based on this, several designs were generated at AiResearch in order to define the most appropriate approach. Figure 3-55 represents the design adapted for future analysis. This design too was modified during the final design layout effort.

17. Electro Magnetic Interference Compatibility

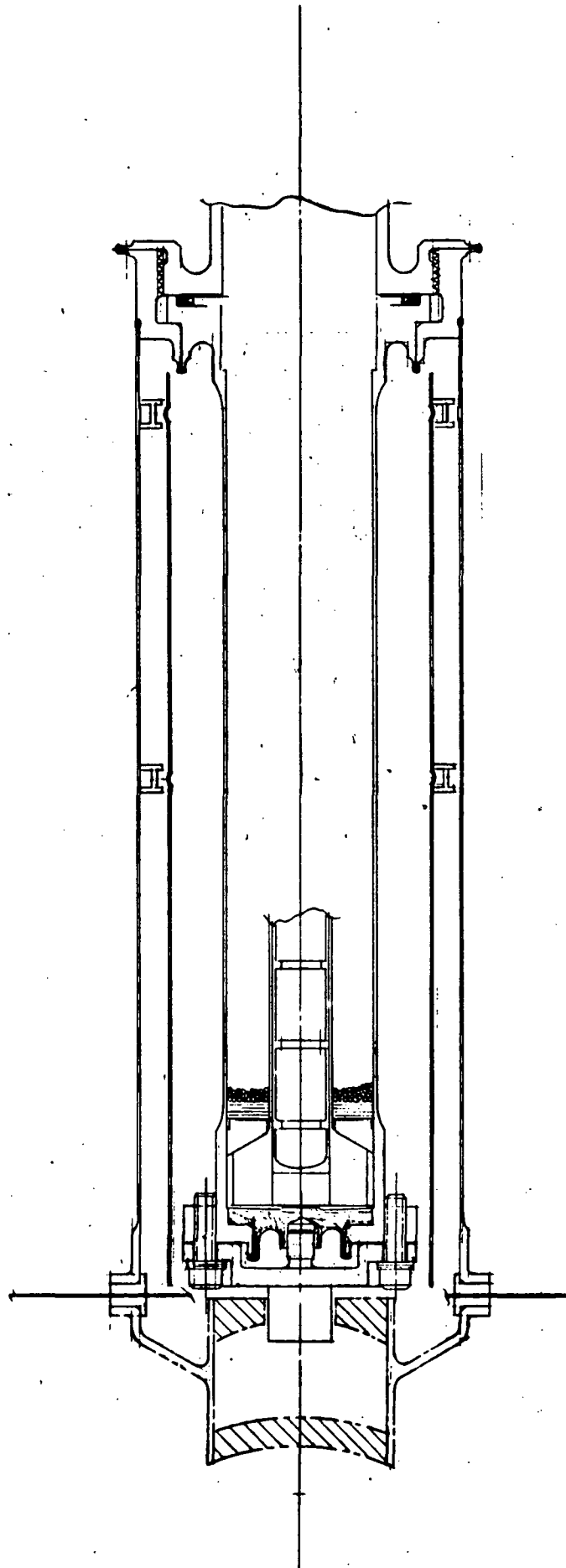
Considerable concern was expressed by HRC regarding the EMI compatibility of the VM refrigerator. A specification and test method was provided to AiResearch, portions of which are included herein for reference: "The EMI levels within the dewar IR detector assembly are not to exceed 0.06 microvolt RMS in any 15 sec period, nor are spikes exceeding 0.2 microvolt P-P to occur more than 1 percent of this period. The frequency spectrum within which these measurements are to be made is to extend from 10 Hz to 155 Hz. The EMI levels outside the dewar are not to exceed 0.6 microvolts in a 15 sec period within the above spectrum nor are spikes exceeding 2.0 micro volts to occur more than 1 percent of this period.

The test methods are specified in MIL-STD-462 as applicable. Methods for bands below 10K Hz are to be submitted for approval. The probe is to consist of a single loop of wire located between the detector mounting position and the lead matrix array at right angles to the centerline of the cold cylinder. A low noise preamp -3db wave optimized for a 20 ohm source is to be connected to the loop."

These requirements were reviewed with the appropriate personnel at AiResearch to determine the impact on the engine design. The following points resulted from this review:

- (a) The motor enclosure is adequate to eliminate any radiated EMI.
- (b) Leads to the motor must be shielded.
- (c) Calrod heater is EMI free although leads must be double shielded from the GSFC supplied controller.
- (d) Test frequencies go down to 30 Hz per MIL-STD-461; the requested requirements would go down to 10 Hz. In their experience, this has never been done at AiResearch or local test facilities. It would require the acquisition of special test equipment in the form of a memory scope for data recording.





S-77567

Figure 3-55. DCA-VM Axial Interface Concept



During the coordination effort with GSFC and HRC, it was concluded that any testing for compliance with the EMI requirements would be accomplished at HRC with the VM integrated into the Radiometer. This is brought about by the unique testing techniques, levels, and instrumentation that is required, and which is available only at HRC. Every area of potential EMI problems will be evaluated and treated accordingly during the design phase of the VM.

18. Mounting of the VM in the Radiometer

During early interface negotiations, HRC presented their desired method of integrating the VM into the radiometer. Figure 3-56 illustrates the requested method. This represented a drastic deviation from the proposed method of integration of the VM since the VM had been designed for two or three point mounting. This technique was utilized in order to render the lightest system possible and to avoid amplification of Radiometer input dynamic loads to the VM.

Three techniques for incorporating the single flange concept of mounting were evaluated and are illustrated in Figures 3-57 through 3-59. Each approach required the use of an exoskeleton or significant reinforcement of the mounting flange systems in order to provide a system stiff enough to allow the VM to function in a normal manner.

After careful consideration of the numerous alternatives available, it was mutually agreed to utilize a two point pickup of the VM at the radiometer. This technique provided the least weight with the maximum package structural rigidity. Figure 3-60 illustrates the method of interfacing the second mount point that has been adopted.

19. Cold End Motion

During the April 11, 1972, interface coordination meeting at GSFC, it was determined that it would be necessary to establish the relative motion of the VM cold end with respect to the mounting flange interface. This information was deemed necessary in order to define whether the Honeywell sensor chip could be integrated directly into the AiResearch VM cold end or whether it would be necessary to devise an isolated platform between the VM and the sensor.

GSFC requested the use of their 5 watt VM (which was undergoing performance testing at AiResearch) for measuring the relative motion, rather than relying solely upon calculated approximations. Several methods of instrumentation were evaluated, including the following final candidates:

- (1) Mechanical indicating methods
- (2) Accelerometers
- (3) Proximity probe indicators



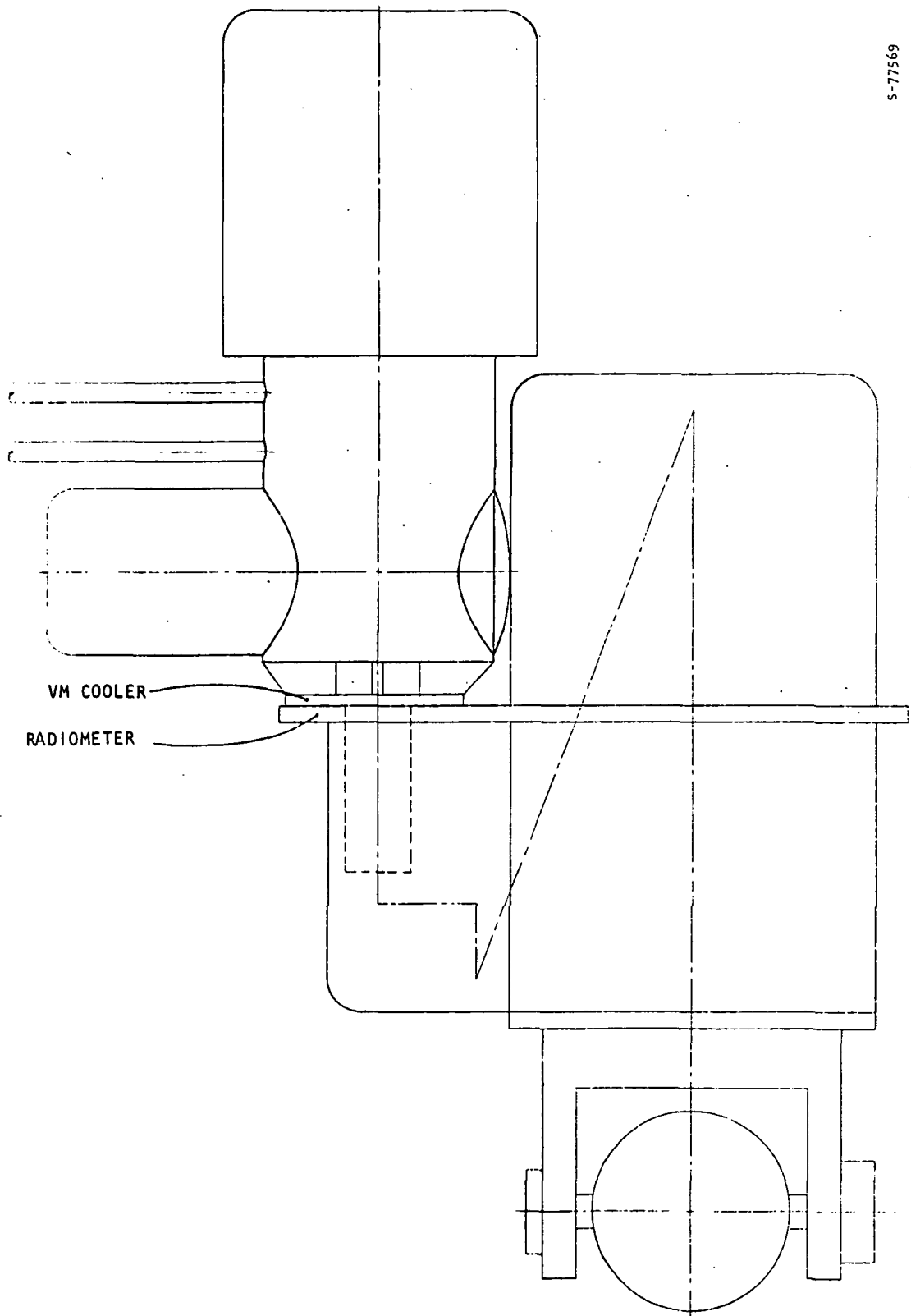
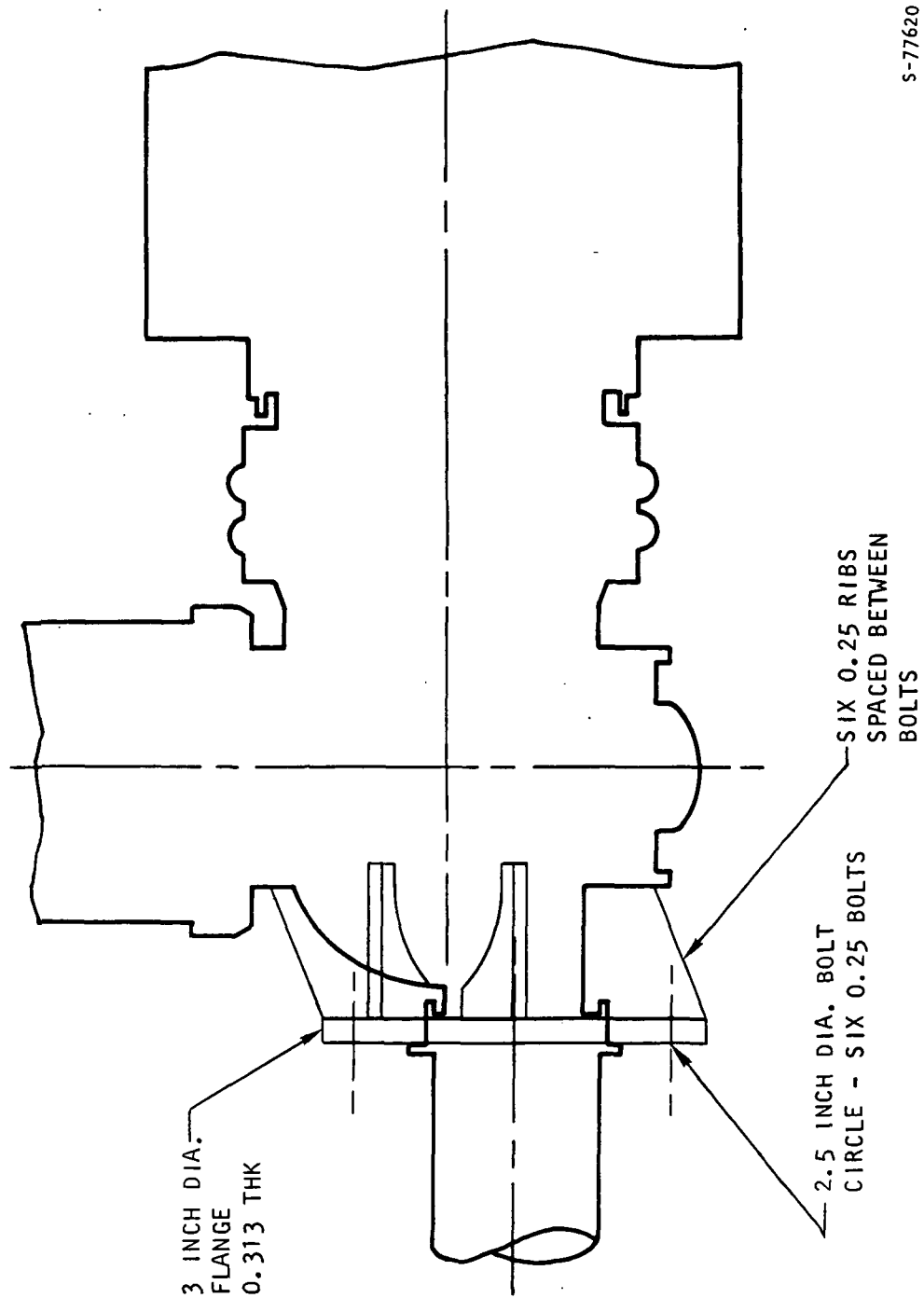


Figure 3-56. HRC Proposed Radiometer-VM Interface

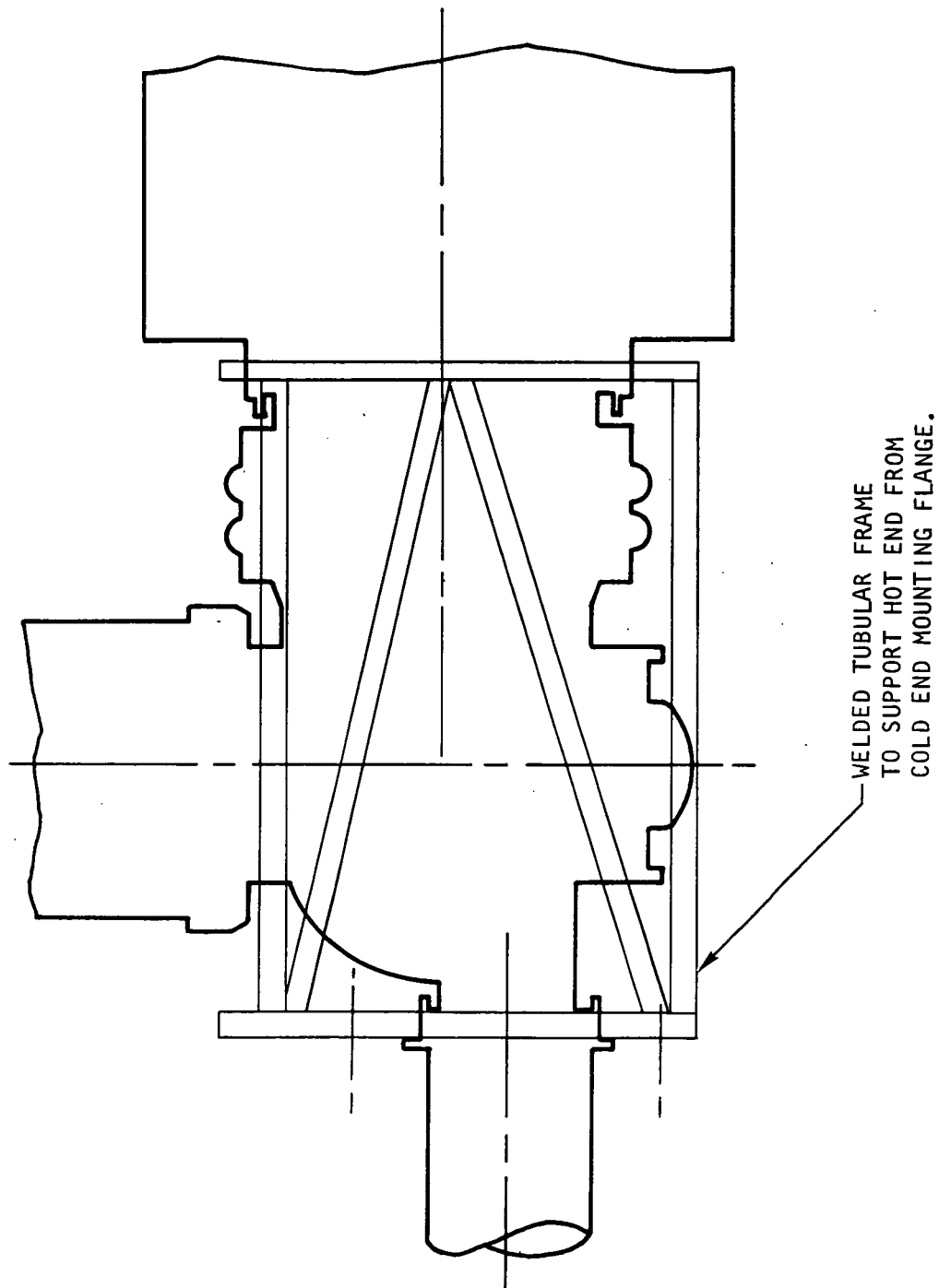




S-77620

NOTE: OUTER SHELL MUST BE INCREASED AS NECESSARY
TO ELIMINATE DEFLECTIONS THAT COULD CAUSE
BEARING PROBLEMS.

Figure 3-57. Flanged Support



S-77619

Figure 3-58. Tubular Frame Support

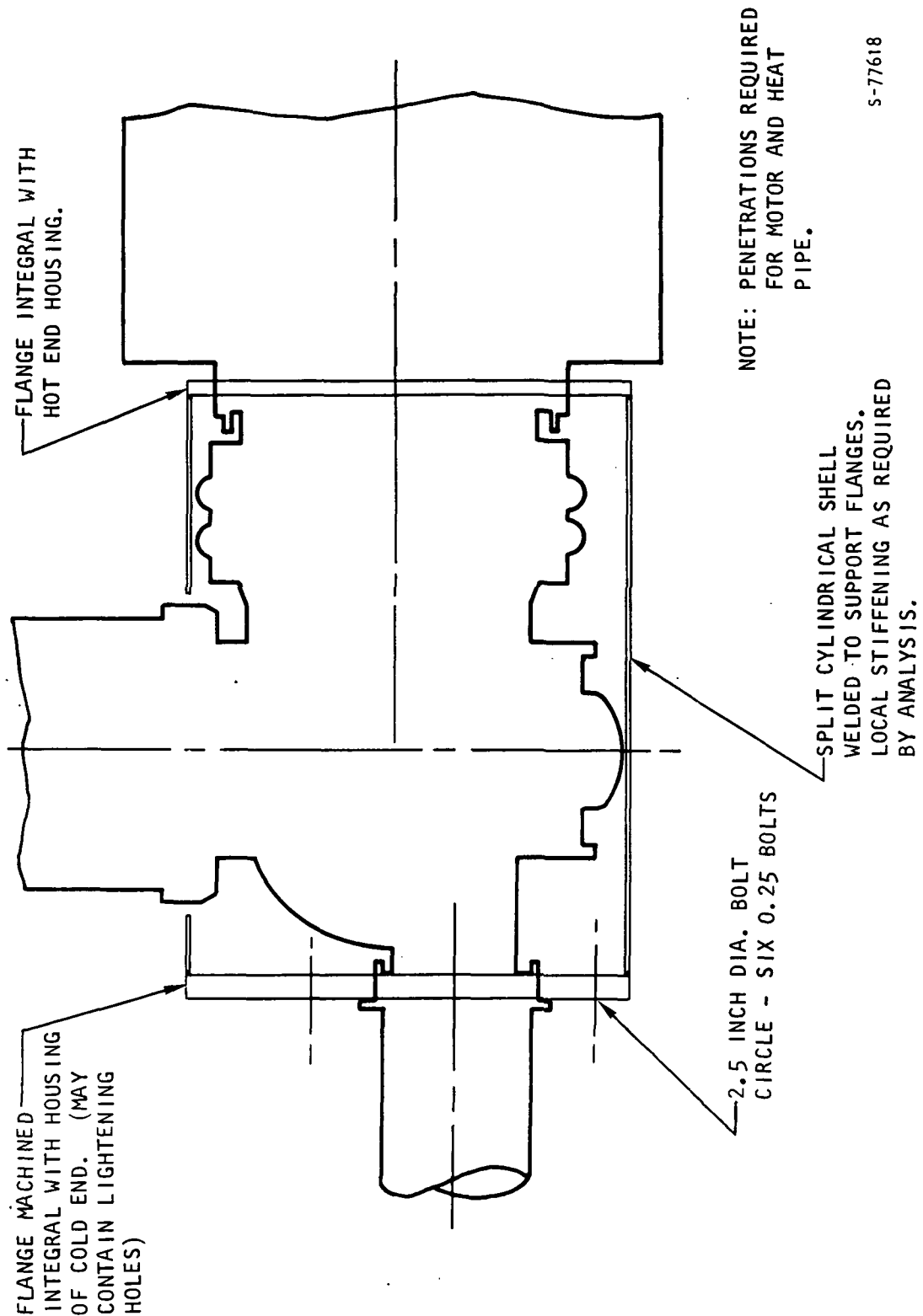
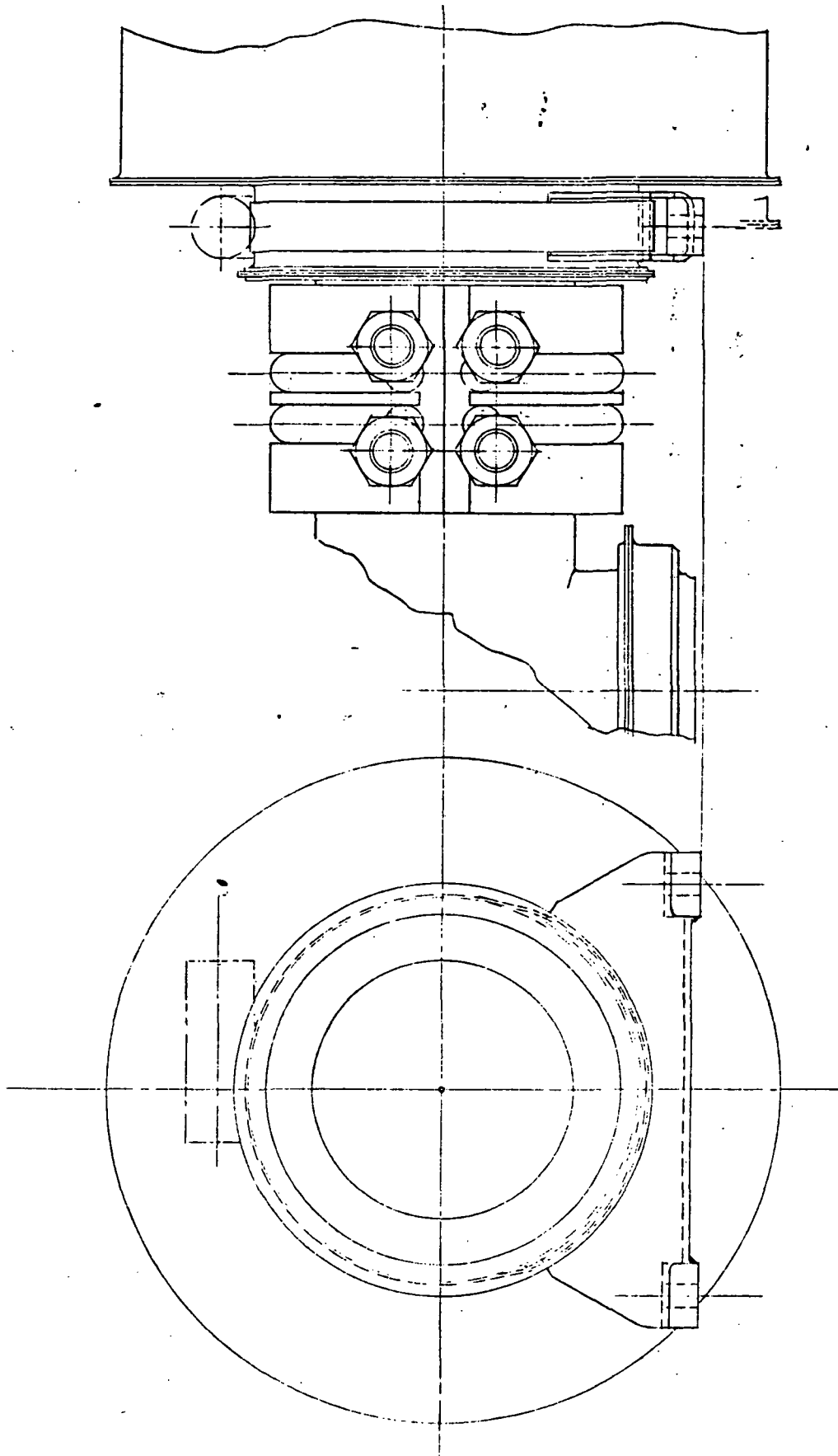


Figure 3-59. Cylindrical Shell Support



S-77564

Figure 3-60. Hot End Mounting Configuration



AIRESEARCH MANUFACTURING COMPANY
Torrance, California

It was determined that the most accurate method would be the use of the proximity coils. These devices inherently have the capability of measuring position deviation on the order of 25 millionths of an inch and have very high response and resolution characteristics.

Figure 3-61 illustrates the instruments which were installed on the GSFC 5 watt VM cold flange. This test setup was to provide two radial modes (90° to one another) and one axial pickup.

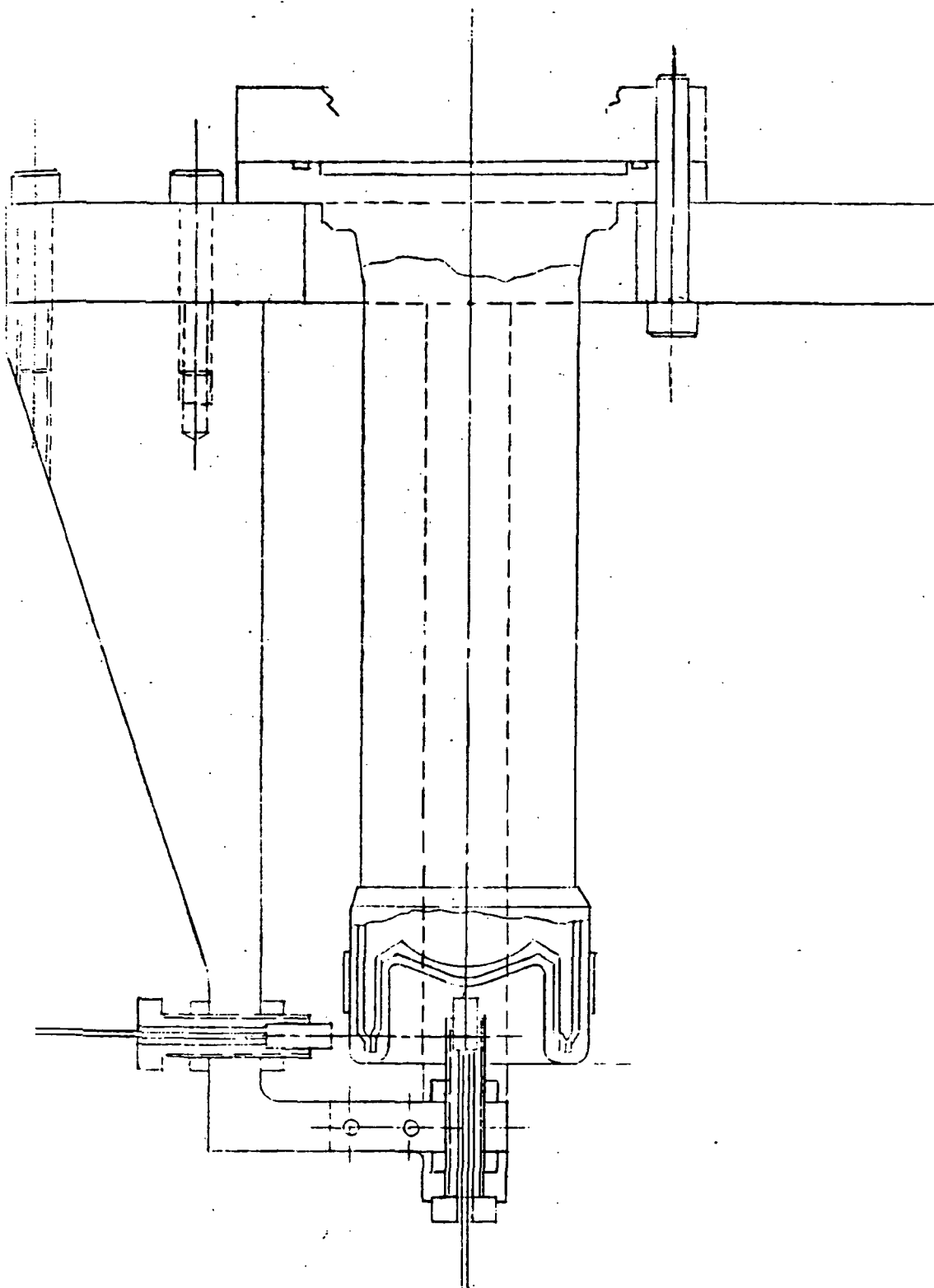
In order to install the sensor system, it was necessary to remove the vacuum dewar jacket, and the multilayer superinsulation. All thermometry instrumentation and load heater leads were maintained intact to eliminate possible performance deviations during subsequent performance evaluation tests. Prior to installing the proximity probes in the holding device, each probe was individually calibrated utilizing a precision micrometer capable of accurate readout to 50 millionths of an inch. The calibration test setup is shown in Figure 3-62. The curves generated by the X-Y plotter are illustrated in Figures 3-63 and 3-64. Figure 3-65 depicts the entire instrumentation system as it was installed.

During the running of the vibration tests it was found that the dimensional deviation from nominal was so small that only a small portion of the calibration curve was useable. Each curve was examined in order to obtain the most linear portion and expanded to produce the curve presented in Figures 3-66 and 3-67. The engine was then run at 300 rpm and the resulting motion recorded on an oscilloscope via the proximity probes. The oscilloscope traces were photographed and are reproduced in Figure 3-68 for evaluation. Prior to starting the rotation of the VM engine, a zero speed or background noise measurement was taken and is also presented in Figure 3-68. From the relative motion noted in the two probes perpendicular to the displacer centerline it was determined that a deviation of $\pm 1.9 \times 10^{-6}$ inches was noted with one while $\pm 6.5 \times 10^{-7}$ inches was noted with the second. No relative motion could be sensed with the third probe (axial motion of the pressure cylinder) in as much as no heat was applied to the hot end and therefore there was no pressure variation in the system.

The results of the 5 watt displacement measurements were analyzed and the following conclusions reached.

- (a) The tests were run with the machine operating at 300 rpm but power was not applied to the hot end. The tests indicated a relative displacement of a few millionths of an inch. This is well below the Honeywell described allowable displacement (by 2 orders of magnitude) and thus problems are not anticipated for the fractional watt refrigerator.
- (b) The test results on the 5 watt VM cannot be considered completely conclusive with respect to the fractional watt machine since the size is different and the tests were not conducted at operating temperatures and corresponding internal pressure cycling.



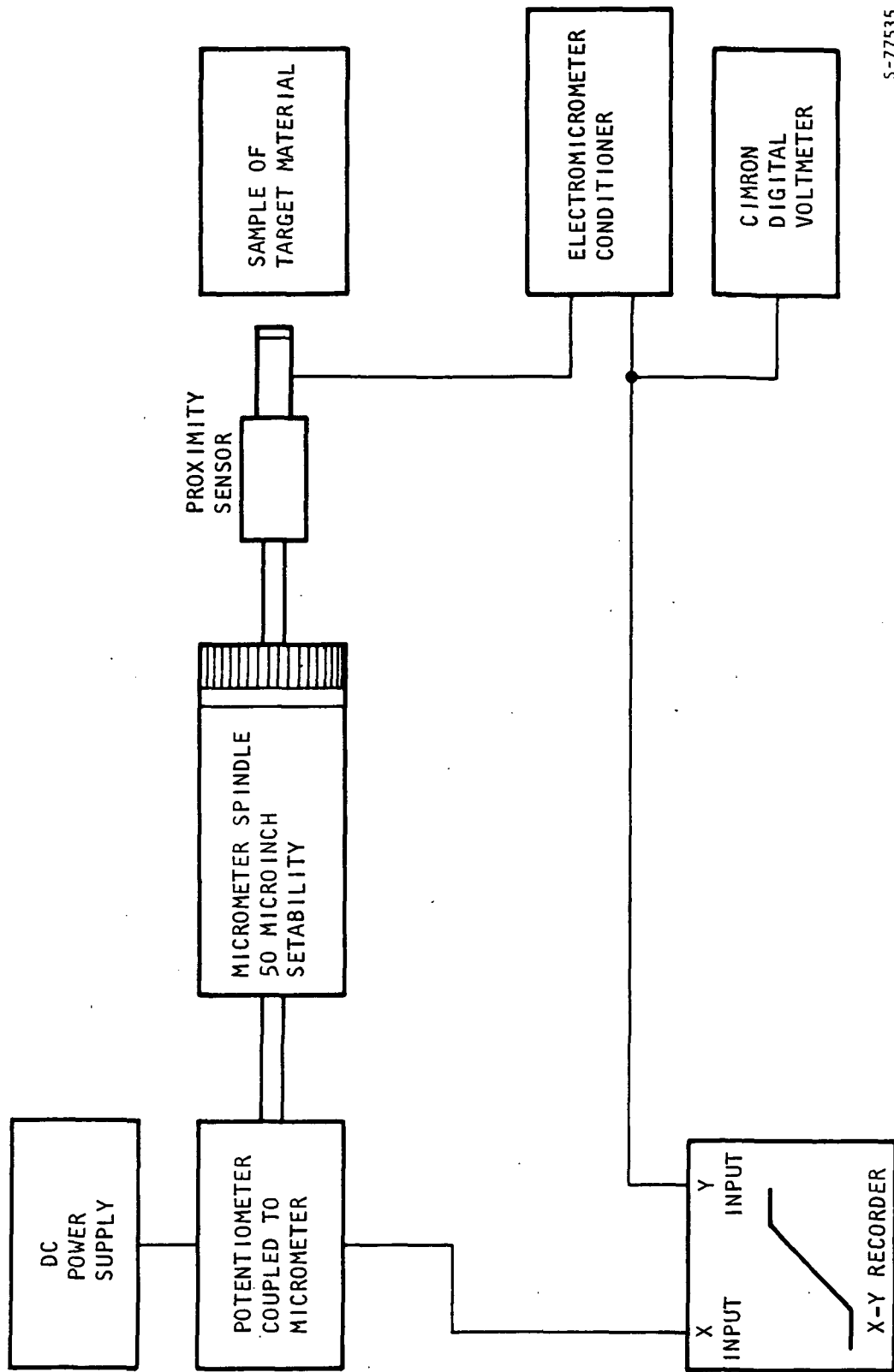


S-77590

Figure 3-16 Figure 3-61. Instrumentation Setup



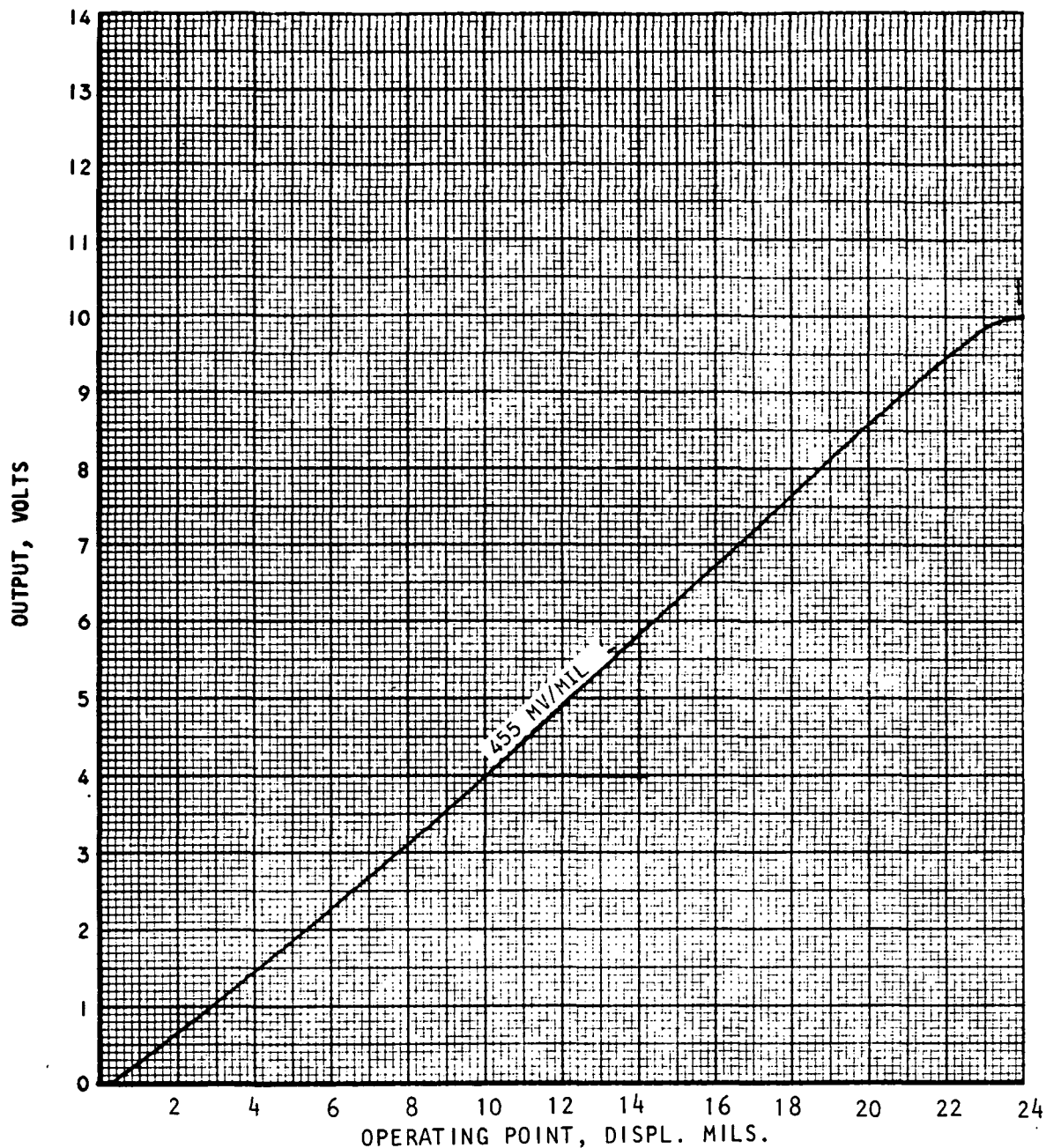
AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California



S-77535

Figure 3-62. Vuilleumier Engine Proximity Sensor System Calibration Test Setup





COIL TYPE 306-2412-02 COIL S/N 73903
 ELECTRO-MIC. S/N 41H450 LINEAR "R" 100 Ω
 SLOPE "R" 51K Ω MATERIAL CU (SIDE)
 CABLE LENGTH AND CAPACITANCE 30 INCHES DUMMY COIL
 DUMMY CAPACITANCE DATE 5-11-72 ENG. OR TECH. THOMAS

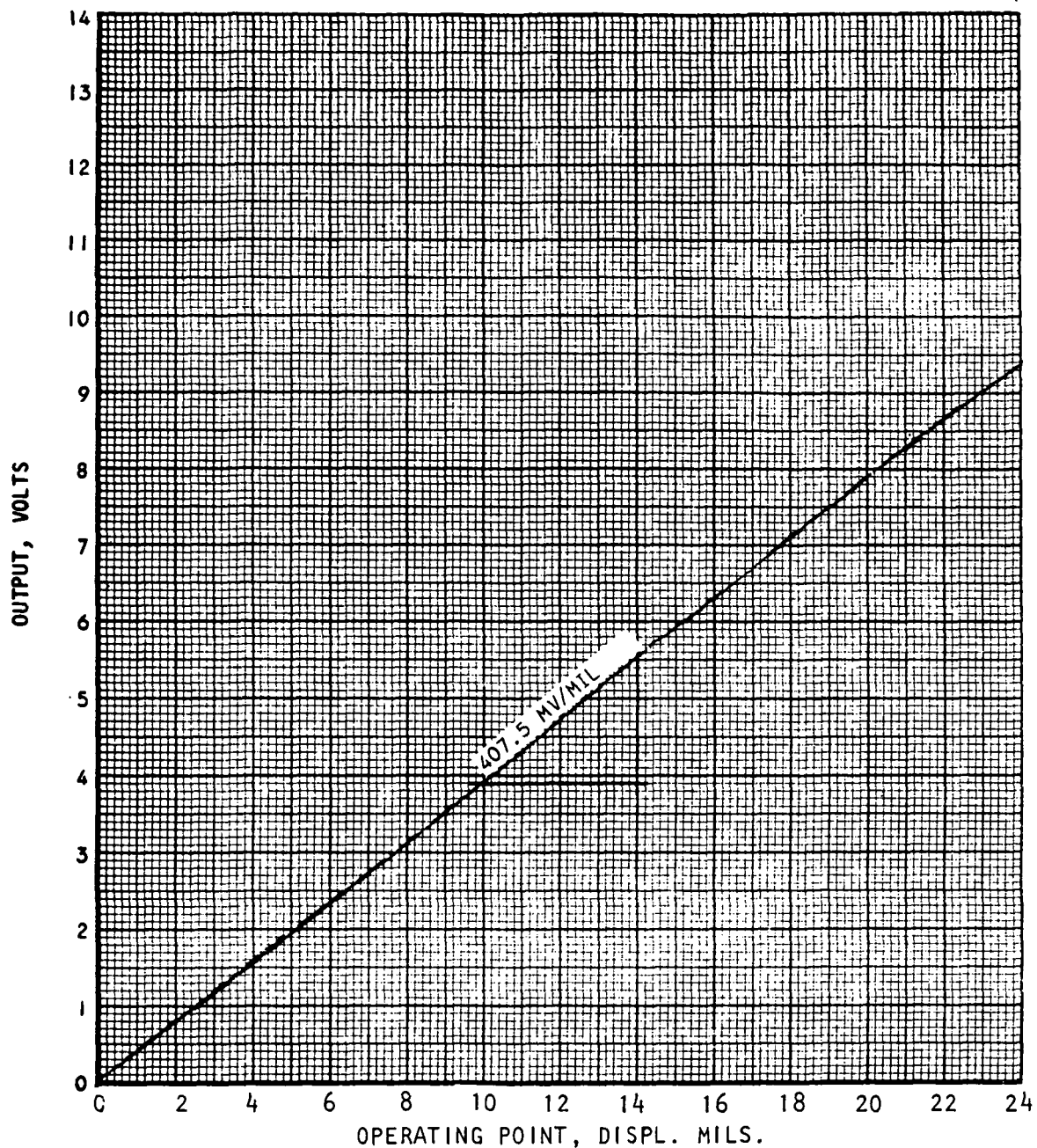
VALIDDYNE - 1.0 G
 3.765 Z

S-77585

Figure 3-63. X-Y Plotter Curve for Probe S/N 41H450



AIRESEARCH MANUFACTURING COMPANY
 Torrance, California



COIL TYPE 306-2412-02 COIL S/N 488310

ELECTRO-MIC. S/N 41X312 LINEAR "R" 110 Ω

SLOPE "R" 47K Ω MATERIAL CU (SIDE)

CABLE LENGTH AND CAPACITANCE 30 INCHES DUMMY COIL _____

DUMMY CAPACITANCE _____ DATE 5-10-72 ENG. OR TECH. THOMAS

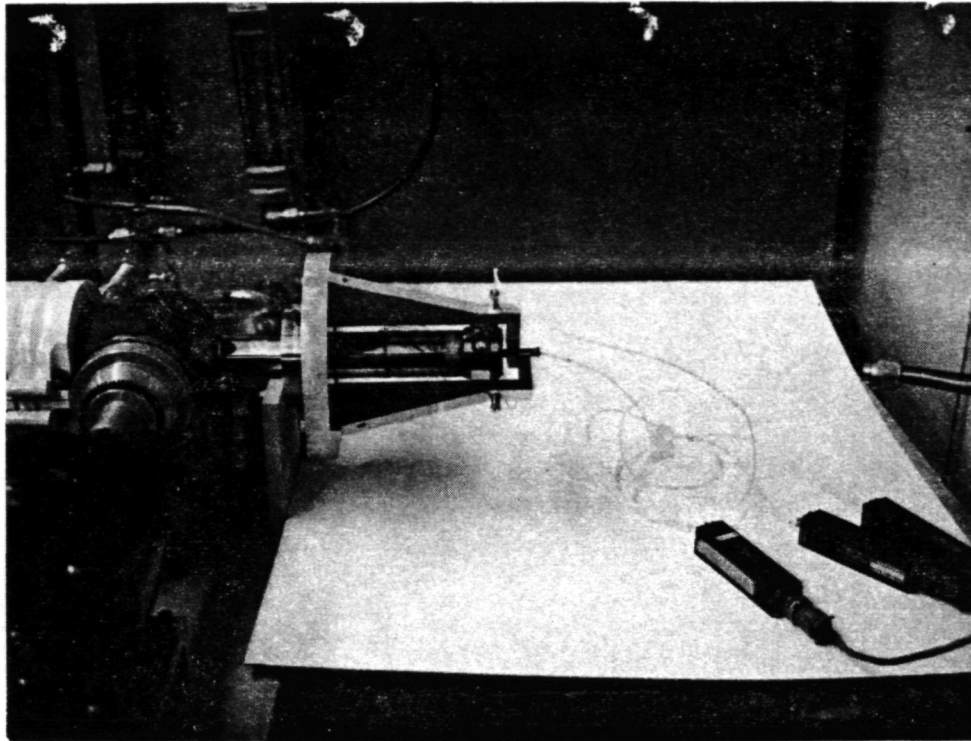
VALIDYNE - GAIN 1.0
ZERO 3.86

S-77584

Figure 3-64. X-Y Plotter Curve for Probe
S/N 41X312



AIRESEARCH MANUFACTURING COMPANY
Torrance, California

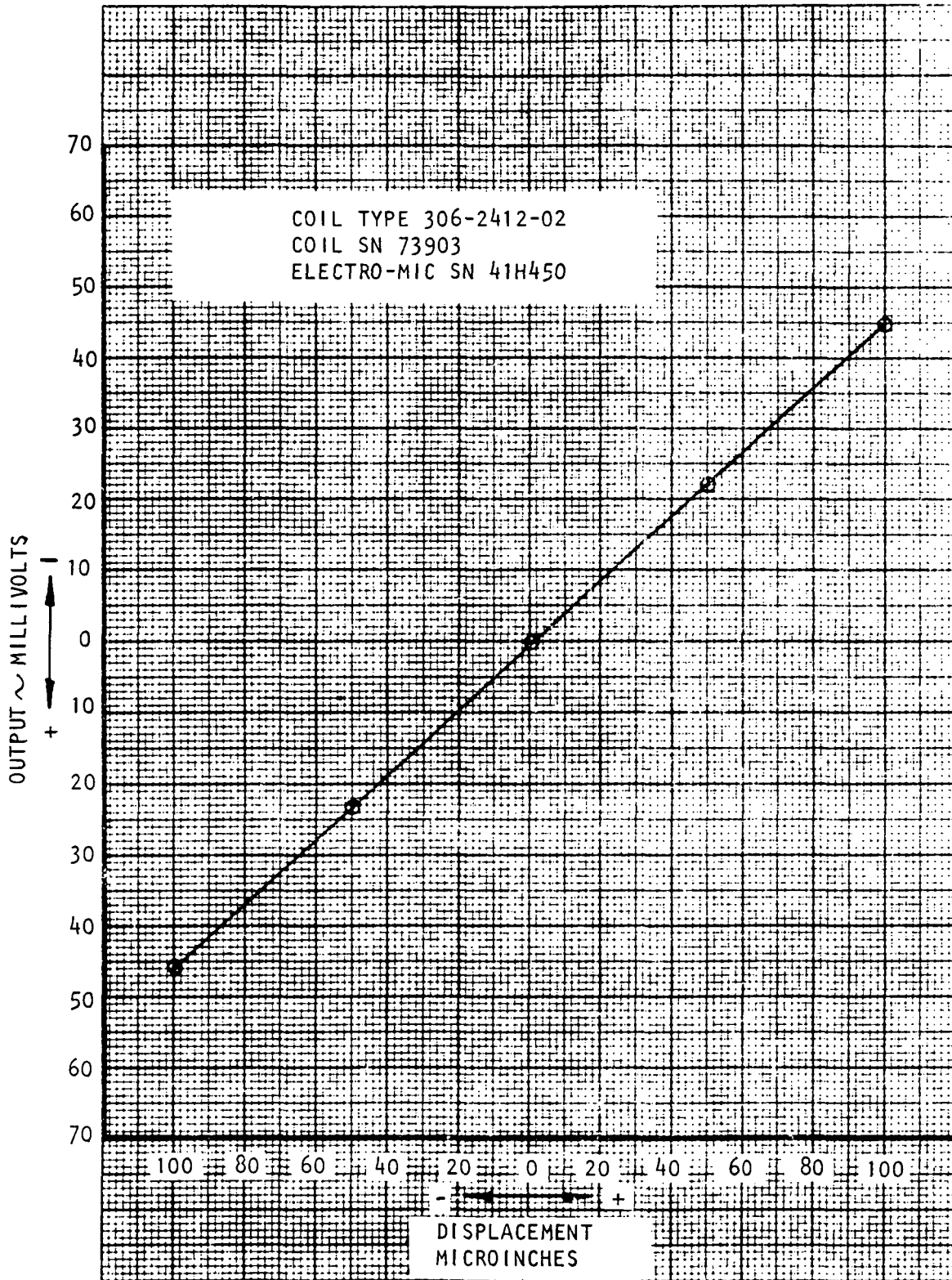


F-16024

Figure 3-65. Proximity Probe Mounting Frame



AIRESEARCH MANUFACTURING COMPANY
Torrance, California

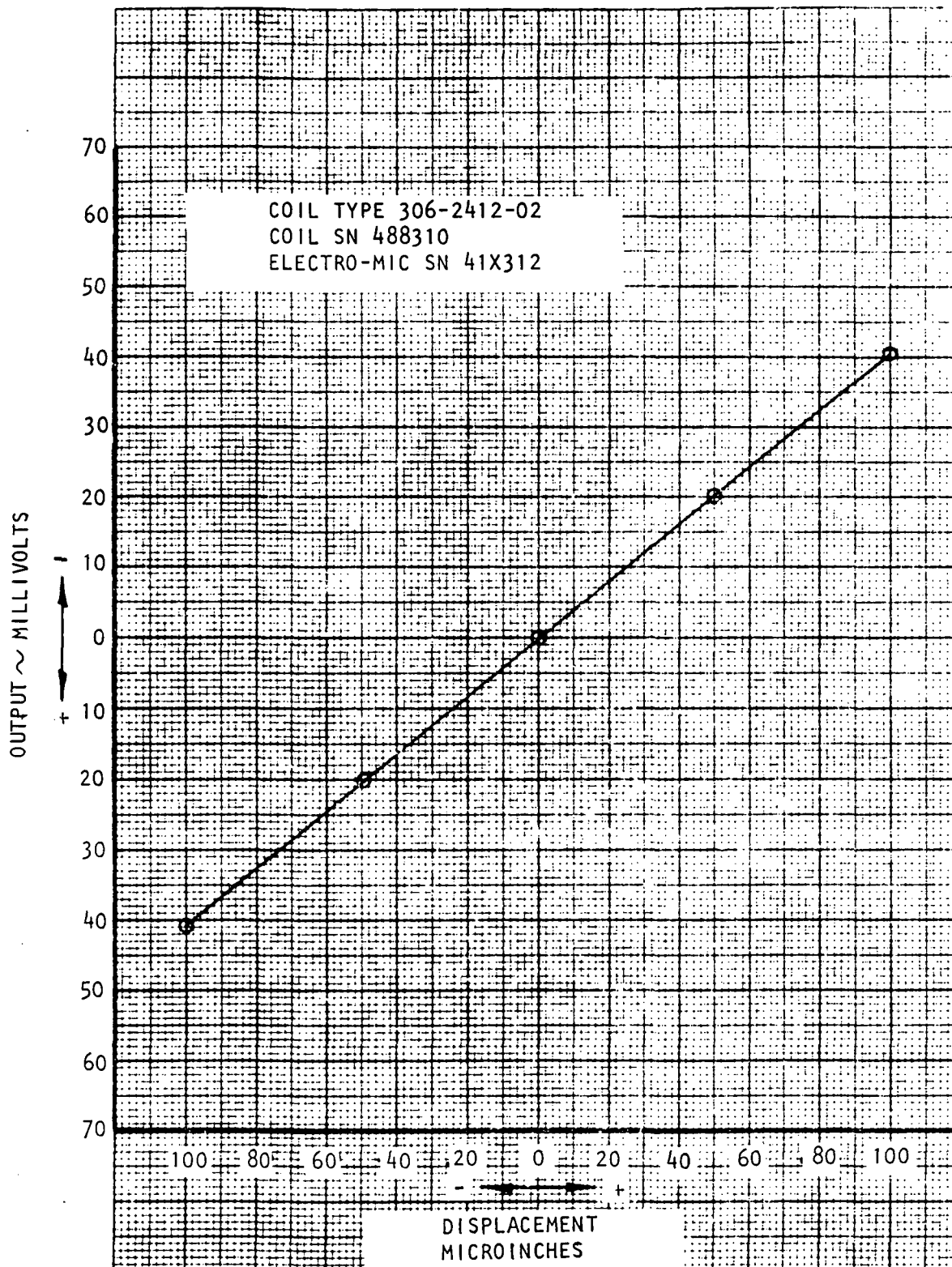


ELECTRO-MICROMETER
CALIBRATION

S-77583

Figure 3-66. Expanded Curve for Probe (SN 41H450)



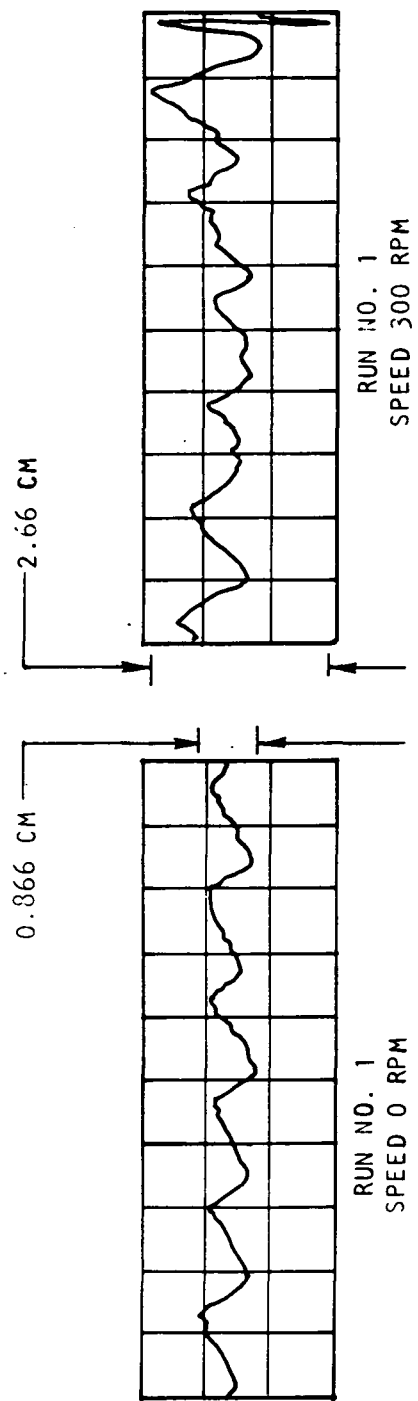


S-77586

ELECTRO-MICROMETER CALIBRATION

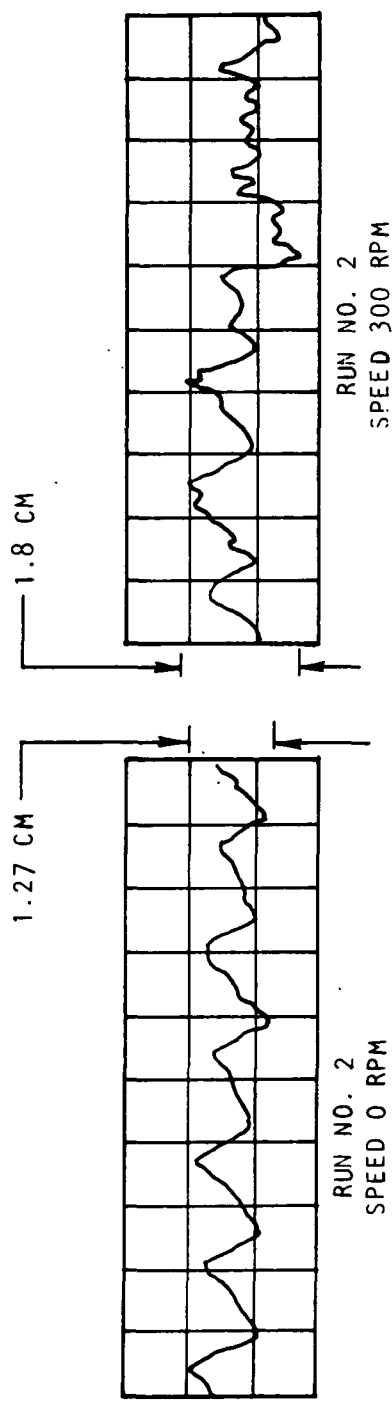
Figure 3-67. Expanded Curve for Probe (SN 41X312)





SWEEP 5 MS/CM
1 MV/CM

SN 414450



SWEEP 5 MS/CM
1 MV/CM

SN 41X312

S-77575

Figure 3-68. Oscilloscope Traces

- (c) Due to (b) above, it is anticipated that the GSFC will require displacement measurements to be made on the fractional watt machine at operating conditions to assure a compatible interface with the Honeywell detector assembly.

Cold End Closure Dome

During the numerous interface meetings with Honeywell, it was determined that the interface for the cold end would exist at the flange provided on the VM cold cylinder for mounting of the Honeywell detector. In addition, the vacuum dewar is to also terminate in a flange compatible with installation of the Honeywell provided lead matrix ring. Both of these interface planes require the design and installation of auxiliary mechanisms if the VM is to be tested at the AiResearch facilities, prior to shipment to the Honeywell Radiation Center. (Integration of HRC equipment into the VM cold end and installation of the VM into the radiometer will occur at Honeywell.) For this reason, two laboratory test devices have been designed to allow operation at AiResearch:

- (1) In order to provide a source of thermal load comparable to that of the Honeywell detector and detector mount base, a dummy detector has been designed incorporating an electrical resistor to simulate the load. This will be installed on the cold heat exchanger at the end of the cold cylinder.
- (2) The vacuum dewar will then be completed by the installation of a temporary closure to allow evacuation of the annulus. This device will be held in place by a ring clamp and utilizes an O-ring seal at the interface to the vacuum dewar cylinder. Provisions for an evacuation tube will be incorporated along with a pinch-off tube cover. With this design, when testing has been completed at AiResearch, the dummy load may be removed, the evacuation closure installed, and the system evacuated and pinched off for shipment. This will protect the cleanliness and efficiency of the superinsulation during transportation and storage.

Figure 3-69 depicts the cold end vacuum dewar test closure and the dummy load interface mechanism.

Final Closure Technique

Early in the design of the fractional watt VM it became clear that it would be necessary to devise a method of closure of the various components that was unique. The closure system requirements were centered around the requirement for a flight weight system while still maintaining the "prototype" capabilities of easy access for removal, replacement, or rework of any of the internal components. With this approach as a baseline, the two chief areas of concern were summarized as follows:



NOTE: EVACUATION TUBE SHOWN IN THE PINCHED
OFF (SHIPPING) CONFIGURATION WITH
PROTECTIVE CAP INSTALLED

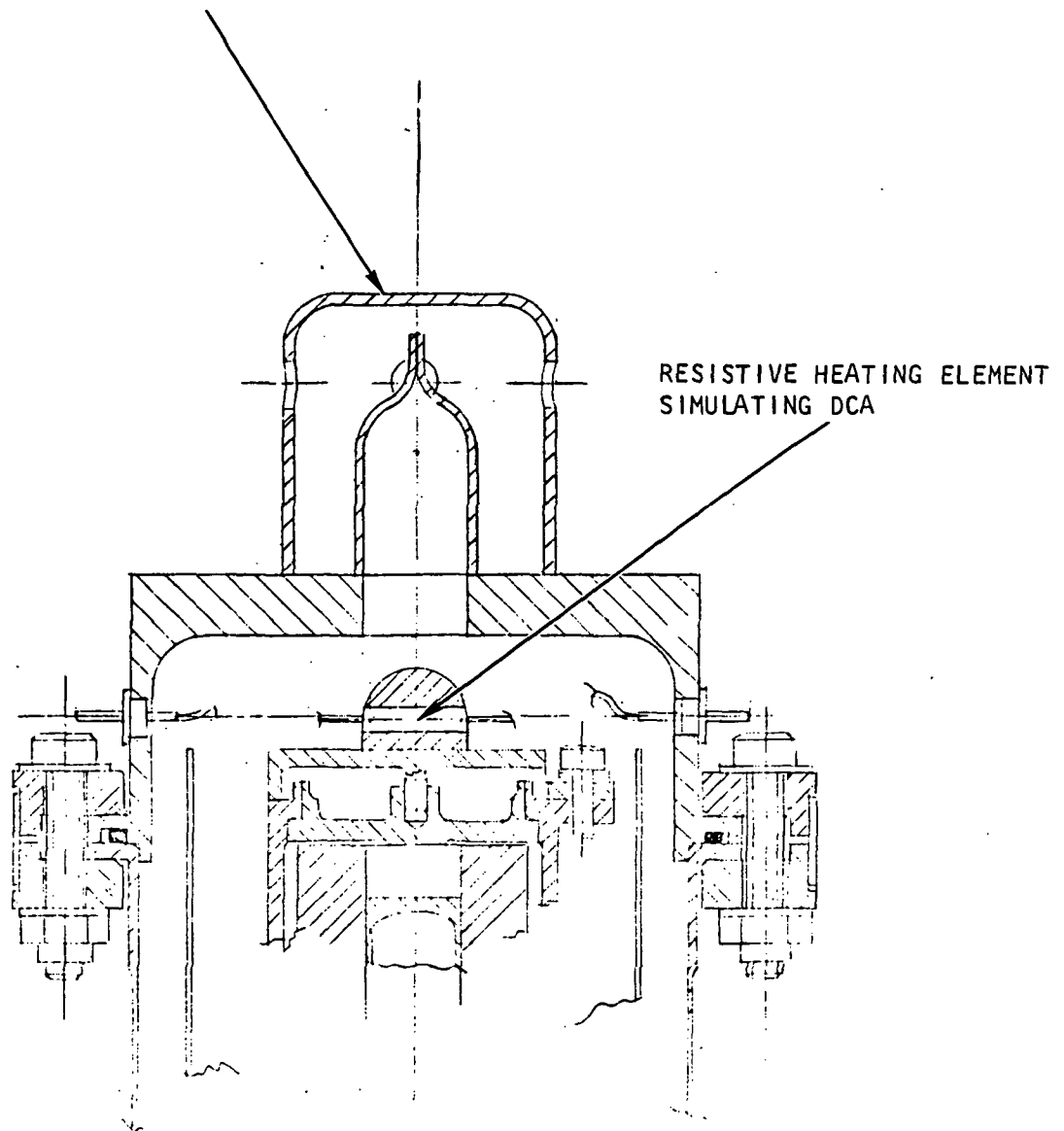


Figure 3-69. Cold End Vacuum Dewar Test and Shipping Closure



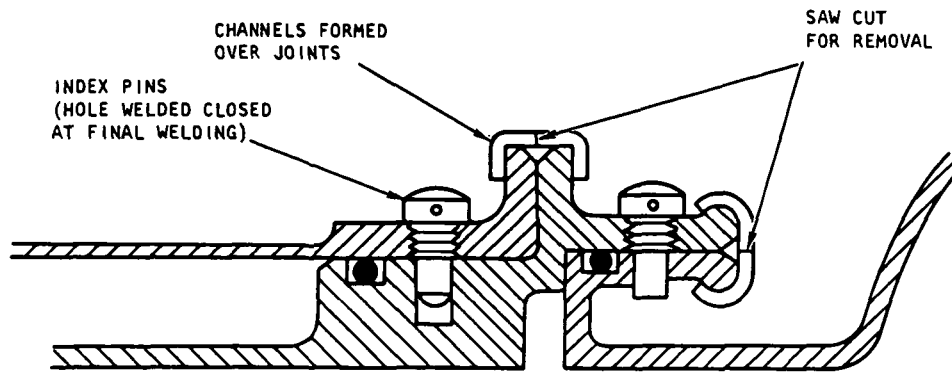
- (1) The joint must be a "bolt together" design to allow easy access to the interior mechanisms while the engine is in the fitup-break in-performance testing phase.
- (2) Once all internal components are correct and the appropriate performance has been achieved, the unit is to be permanently sealed and the bolting mechanisms removed resulting in the lightest design possible. Leakage of the operating fluid is not to exceed approximately 1×10^{-8} scc/sec helium at one atmosphere equivalent differential pressure. In addition, no contamination of the operating fluid is to occur as the result of the retention of any sealing mechanism or as the result of excessive outgassing of the sealing system during any material joining necessary to accomplish the required seal.

Figures 3-70 through 3-81 illustrate several of the concepts generated early in the design phase. In general, fault was found with each approach as shown with each illustration. The use of a breach-lock mechanism employing a double weld joint lead to the solution of the problem.

After several design iterations, a concept for producing the final closure joints was developed. Figures 3-82 and 3-83 illustrate the initial concept as it applies to the three primary joints under consideration: the cold end; hot end, and crankshaft closures. This method utilizes a threaded ring that allows all mechanical and pneumatic loading to be carried by the threads. Two methods of sealing the closure may be utilized, dependent upon the particular mode of fabrication. Initially a "K" seal or metallic O-ring could be utilized allowing easy access to internal components without having to destroy any of the closure mechanism. Following the required bearing and engine break in period, and after all required entry to the interior components has been accomplished, the threaded ring is removed and the sealing mechanism destroyed by punching one edge locally. The threaded ring is then replaced and the two small standing edges are welded. This procedure results in a sealing system wherein all mechanical and pneumatic loads are carried by the threaded ring. The only load imparted to the standing edge welds is the internal pressure acting on the weld members themselves. It should be noted that the metallic seal was punched to assure that the seal was destroyed, thus providing a positive gas path to the welded area during helium mass spectrometer leak check, after welding is complete. It is currently anticipated that three closure tools will be utilized to properly compress the metallic seal so as not to rely upon the threaded ring to pull the two components together.

The principles illustrated in Figures 3-82 and 3-83 were then further refined for closure joints in the sump-crankshaft region (Figure 3-84), the cold end of the engine (Figure 3-85) and the hot regenerator--displacer complex (Figure 3-86). Additional design modifications will be incorporated during the second and third tasks of this contract and as the result of the test program to be conducted during the task two activities. In order to fully define the test sequence and parameters to be evaluated, the test plan for the final closure sealing system has been included as Appendix B of this report.

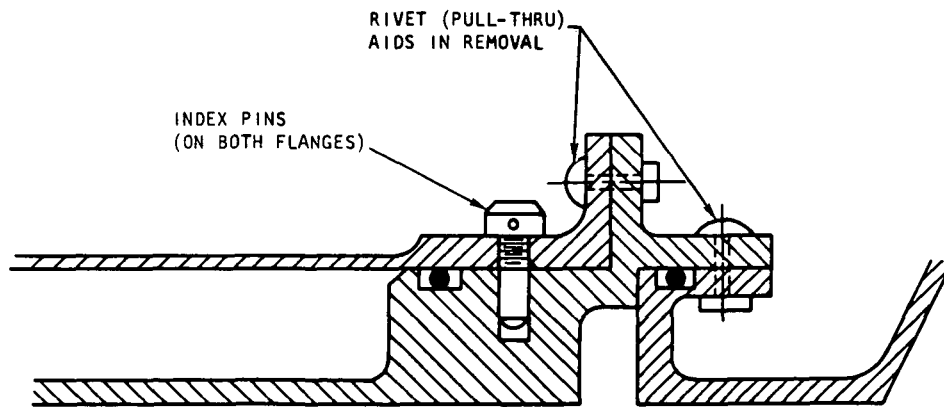




NEGATIVE POINTS:

1. CHANNELS MUST BE CUT OFF THUS CAUSING METAL CHIPS THAT COULD FALL INTO UNIT
2. RELIABILITY OF CLOSED CHANNEL TO TAKE FULL LOAD
3. DIFFICULTY OF FORMING CHANNELS ON COMPLETED ENGINE THAT COULD DAMAGE ENGINE

Figure 3-70. Formed Channel



ALL RIVET HOLES & PIN HOLES MUST BE
WELDED CLOSED AT FINAL WELDING

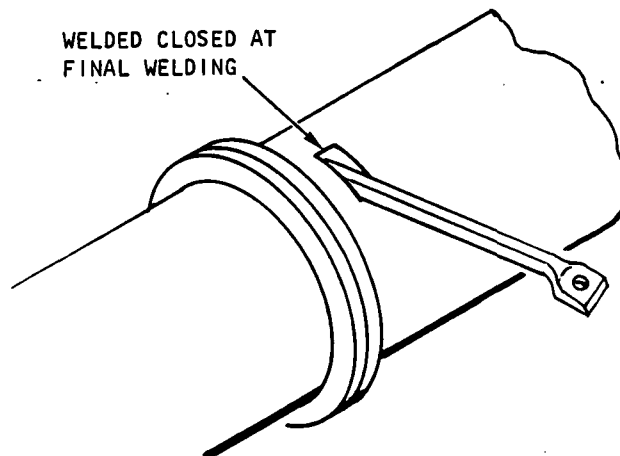
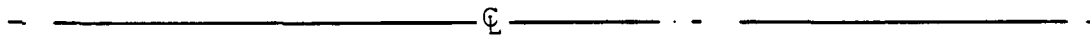
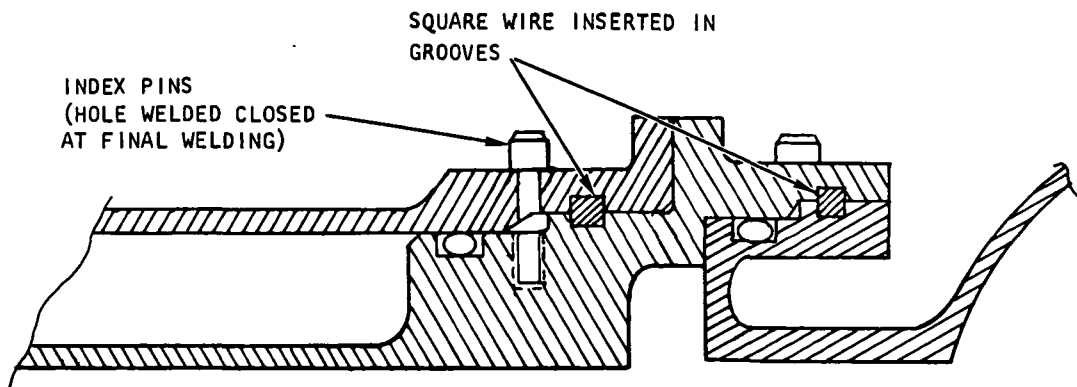
NEGATIVE POINTS:

1. POOR RIVET INSTALLATION ON CURVED SURFACE
2. RIVETS MUST BE DRILLED OUT, THUS CAUSING METAL CHIPS THAT COULD FALL INTO UNIT
3. MUST CARRY FULL LOAD

S-77536

Figure 3-71. Riveted Assembly





S-77562

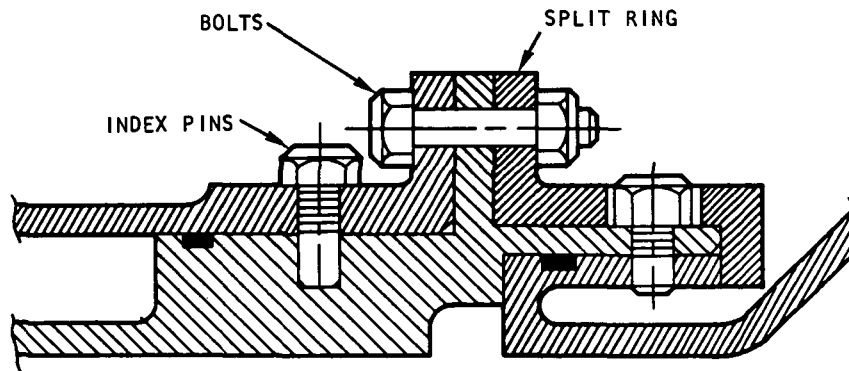
NEGATIVE POINT:

1. POSSIBILITY OF WIRE BREAKING OFF INSIDE

Figure 3-72. Square Wire Retaining Ring

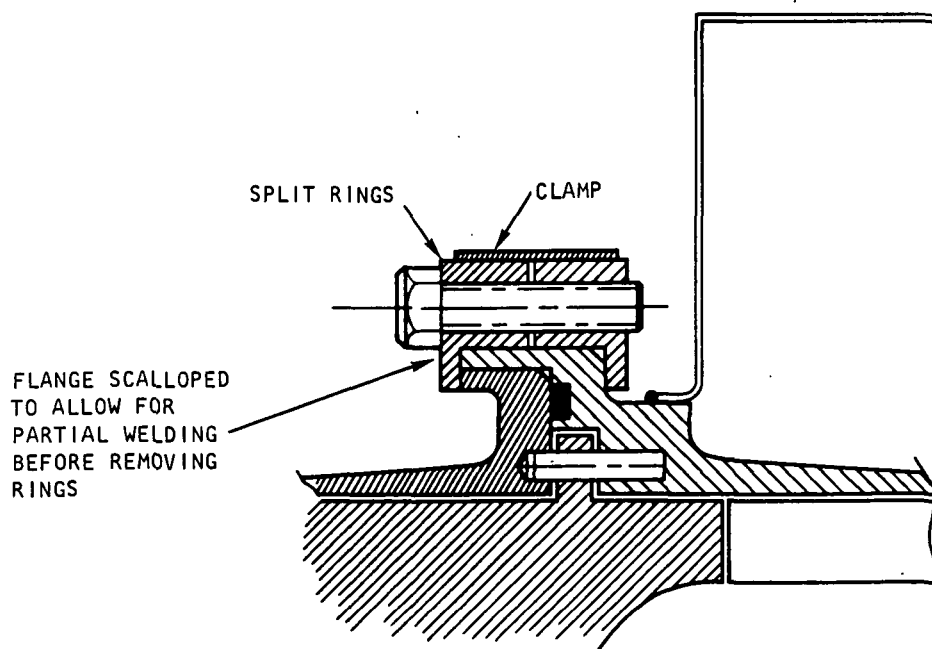


AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California



ALL BOLT HOLES AND INDEX PIN HOLES MUST BE WELDED
CLOSED AT FINAL WELDING.

Figure 3-73. Split Ring Assembled with Bolts

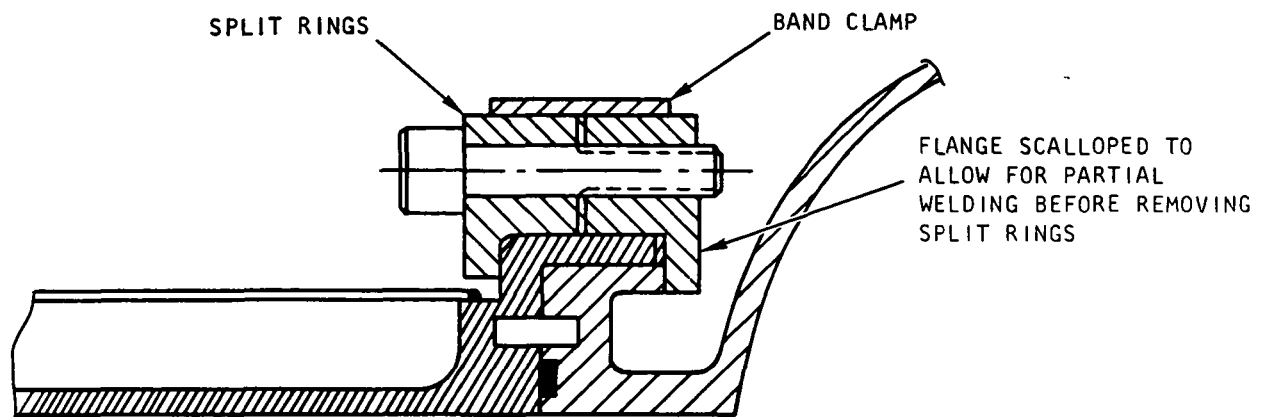


© HOT DISPLACER

S-77557

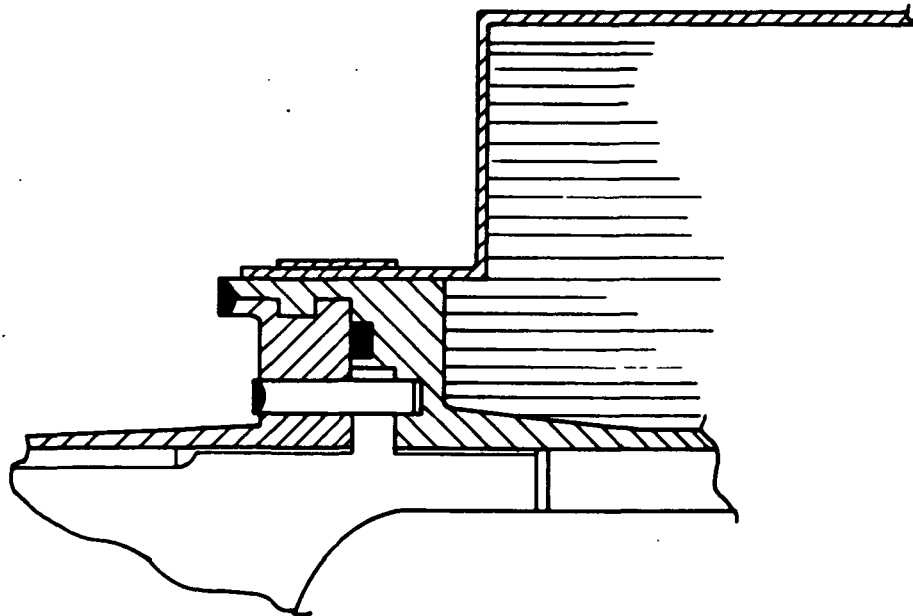
Figure 3-74. Removable Split Ring Fixture -- Hot Displacer
Housing Connection to Sump





— — — — —
 ⌒ COLD DISPLACER

Figure 3-75. Removable Split Ring Fixture--Cold Displacer Vessel Connection to sump



— — — — —
 ⌒ HOT DISPLACER

Figure 3-76. Bayonet Type Lock--Hot Displacer Housing Connection to Sump

S-77552



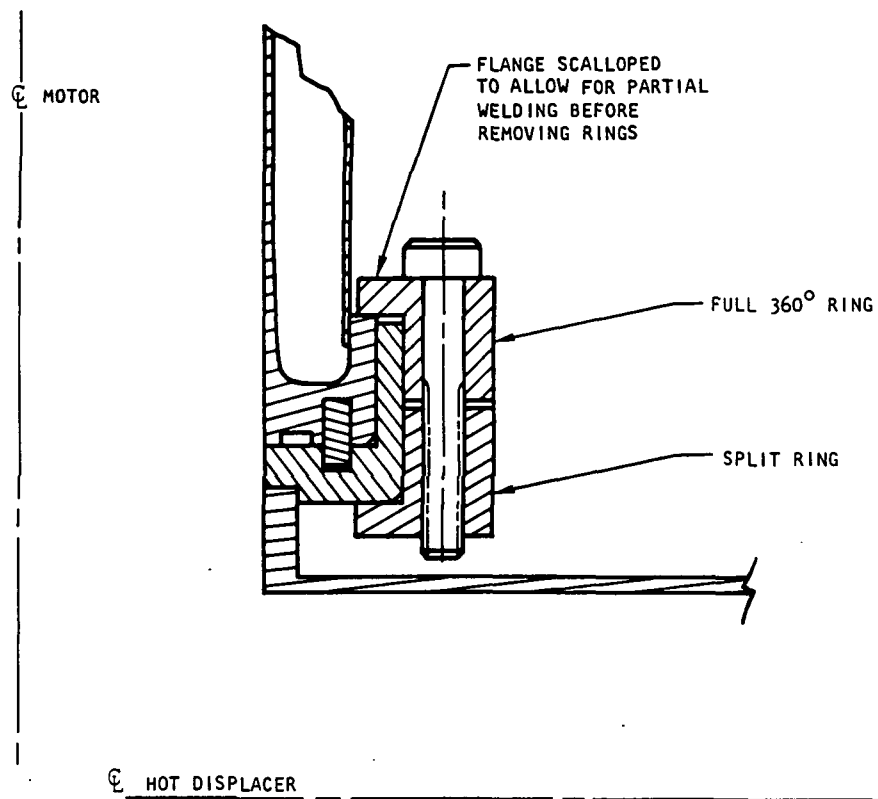


Figure 3-77. Removable Split Ring Fixture Motor Housing .
Connection to Hot Displacer Housing

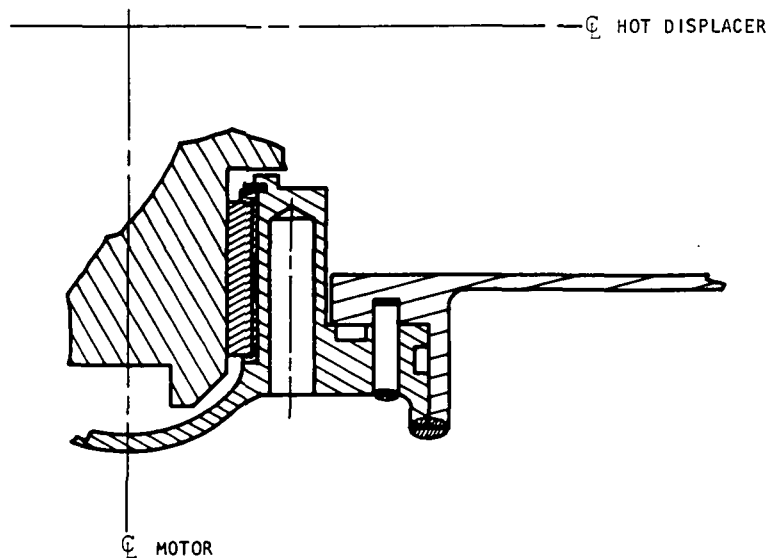


Figure 3-78. Bayonet Type Lock--Crankshaft Closure Dome
Connection to Housing

S-77579



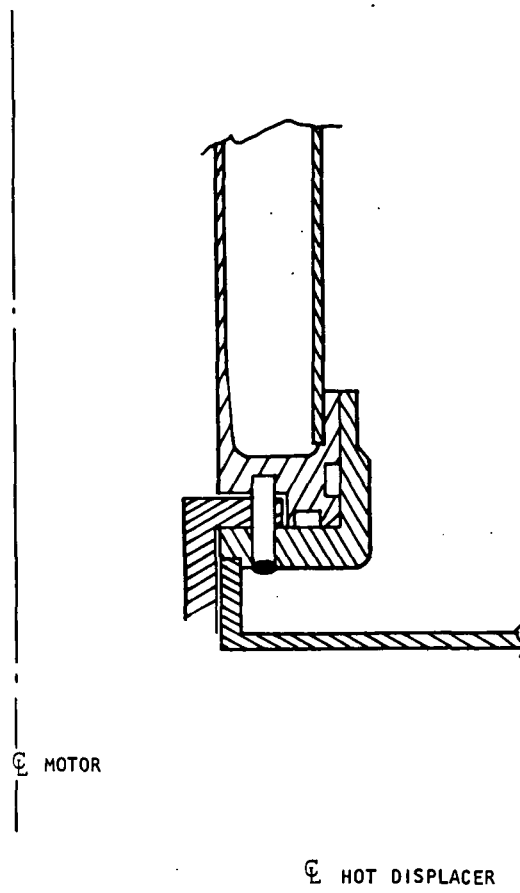


Figure 3-79. Bayonet Type Lock--Motor Housing Connection to Hot Displacer Housing

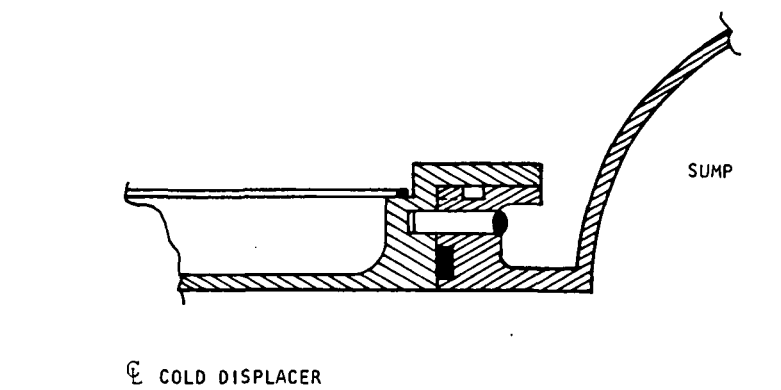
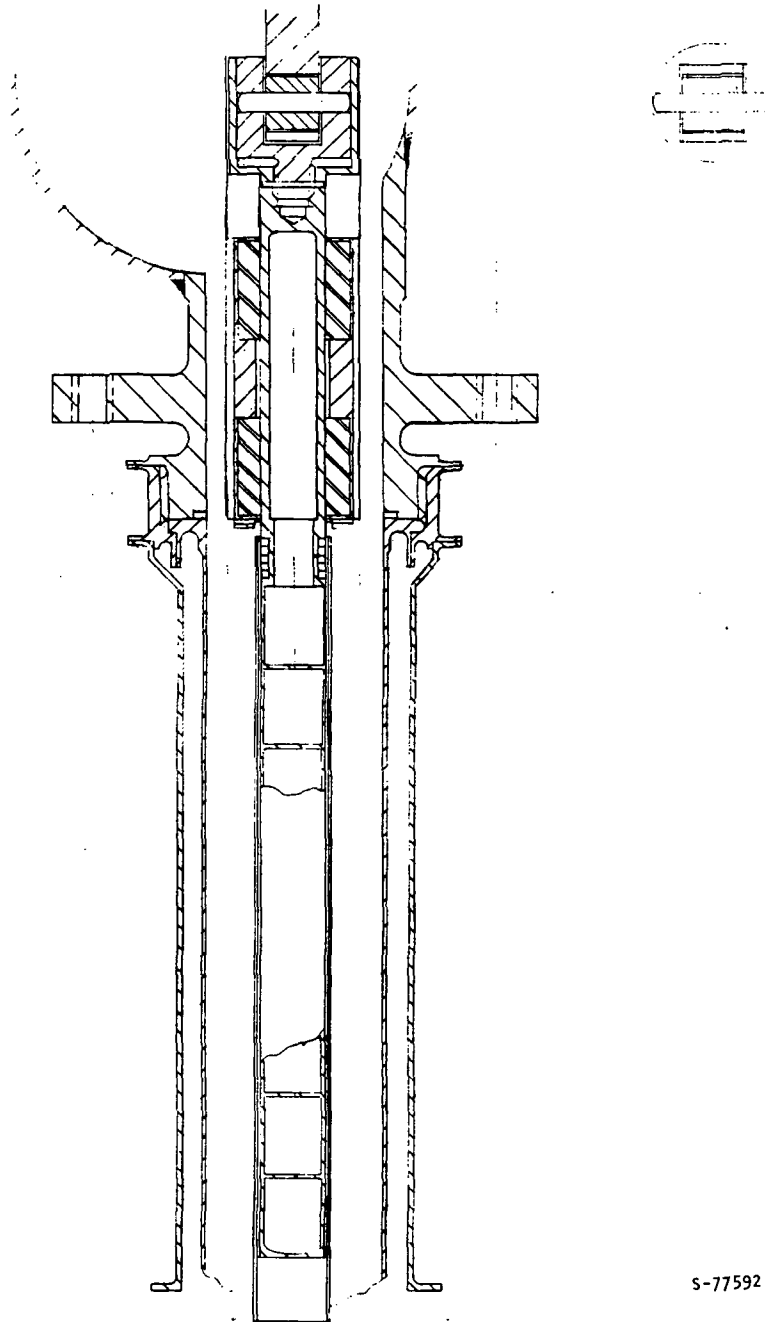


Figure 3-80. Bayonet Type Lock--Cold Displacer Vessel Connection to Sump

S-77578



COLD END CONCEPTS



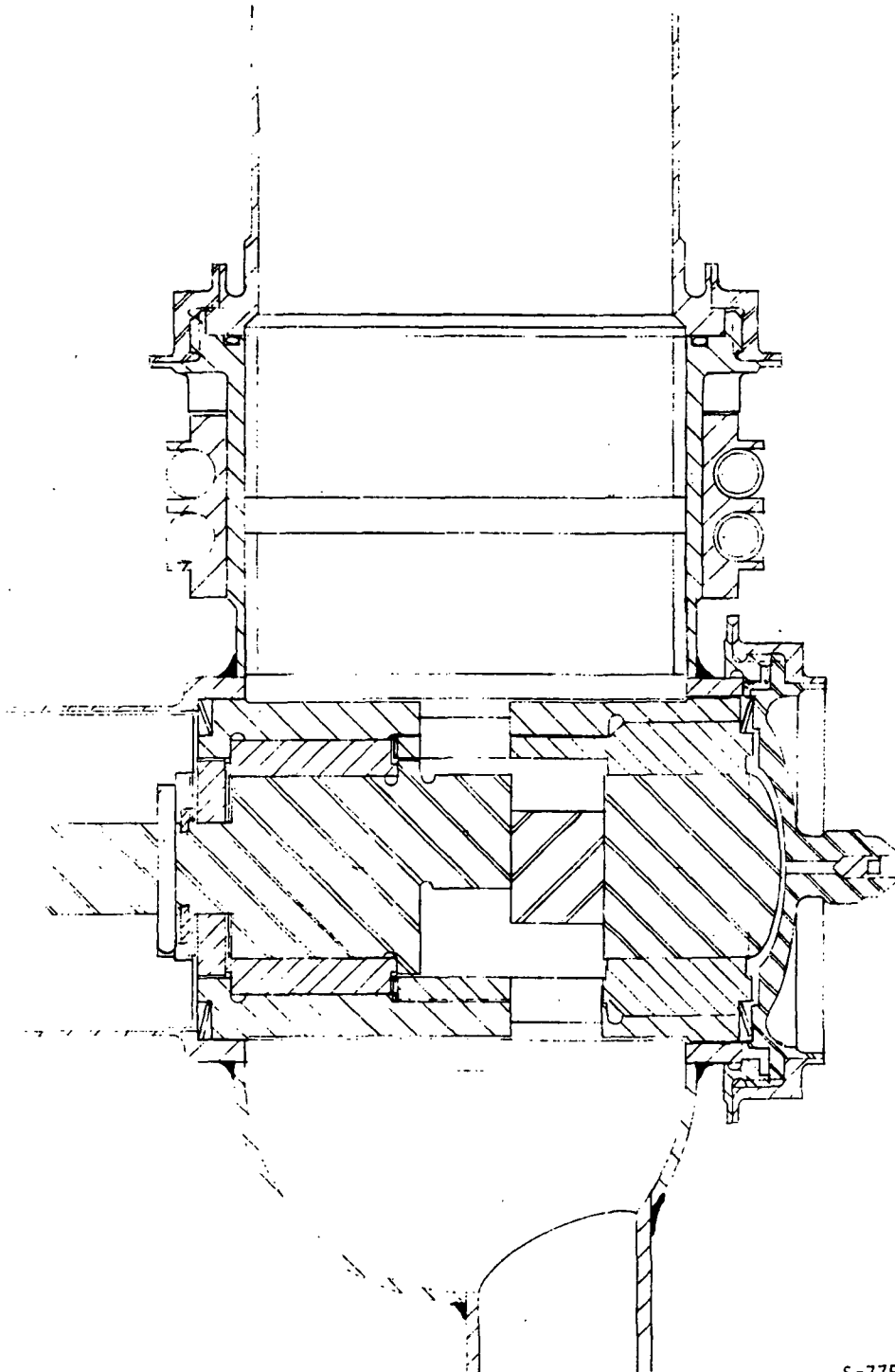
S-77592

Figure 3-82. Cold End Closure Concept



AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California

CONCEPTUAL SUMP
LAYOUT



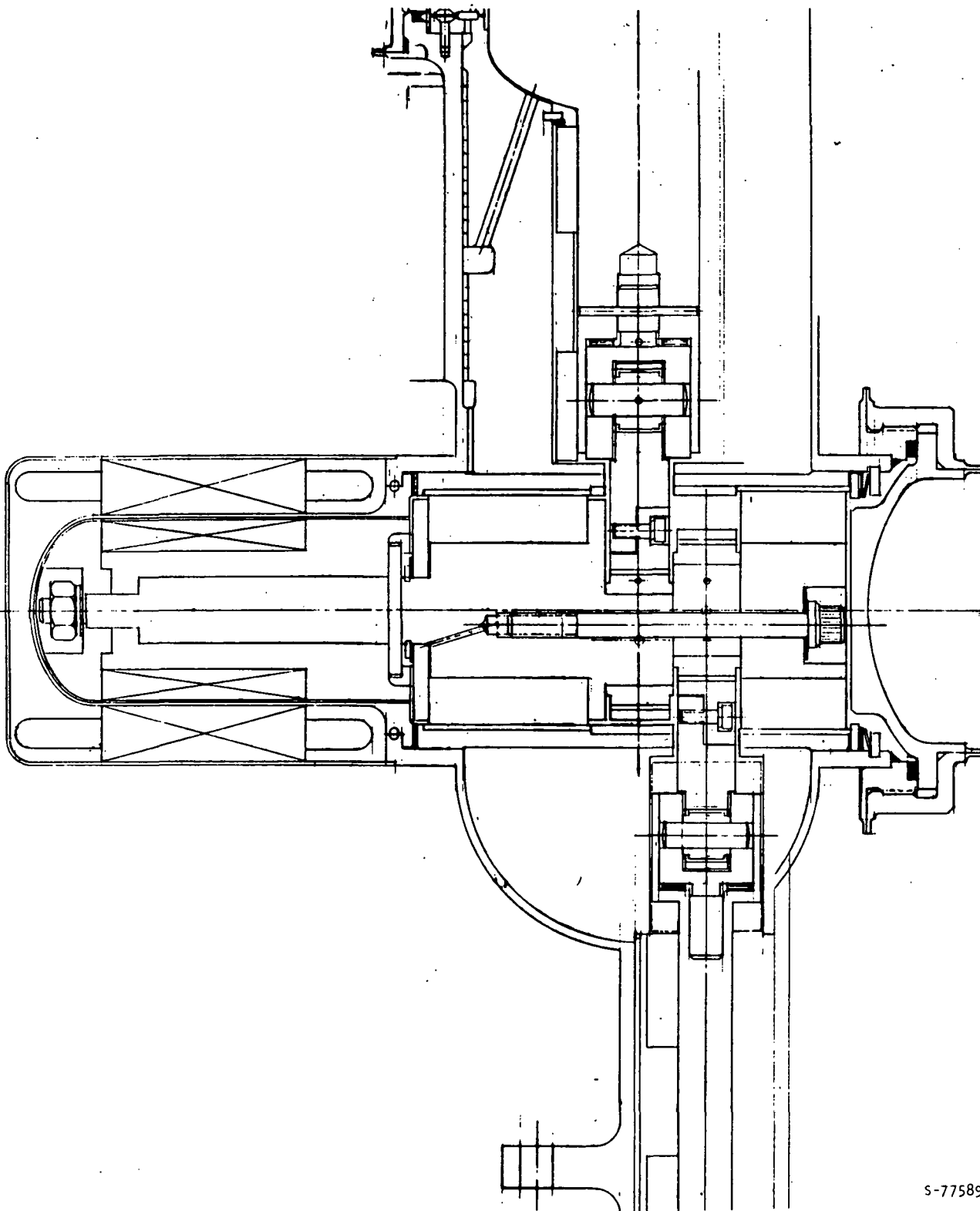
S-77565

Figure 3-83. Hot End and Crankshaft Closure Concept



AIRESEARCH MANUFACTURING COMPANY
Torrance, California

72-8846
Page 3-160



S-77589

Figure 3-84. Sump Preliminary Design



AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California

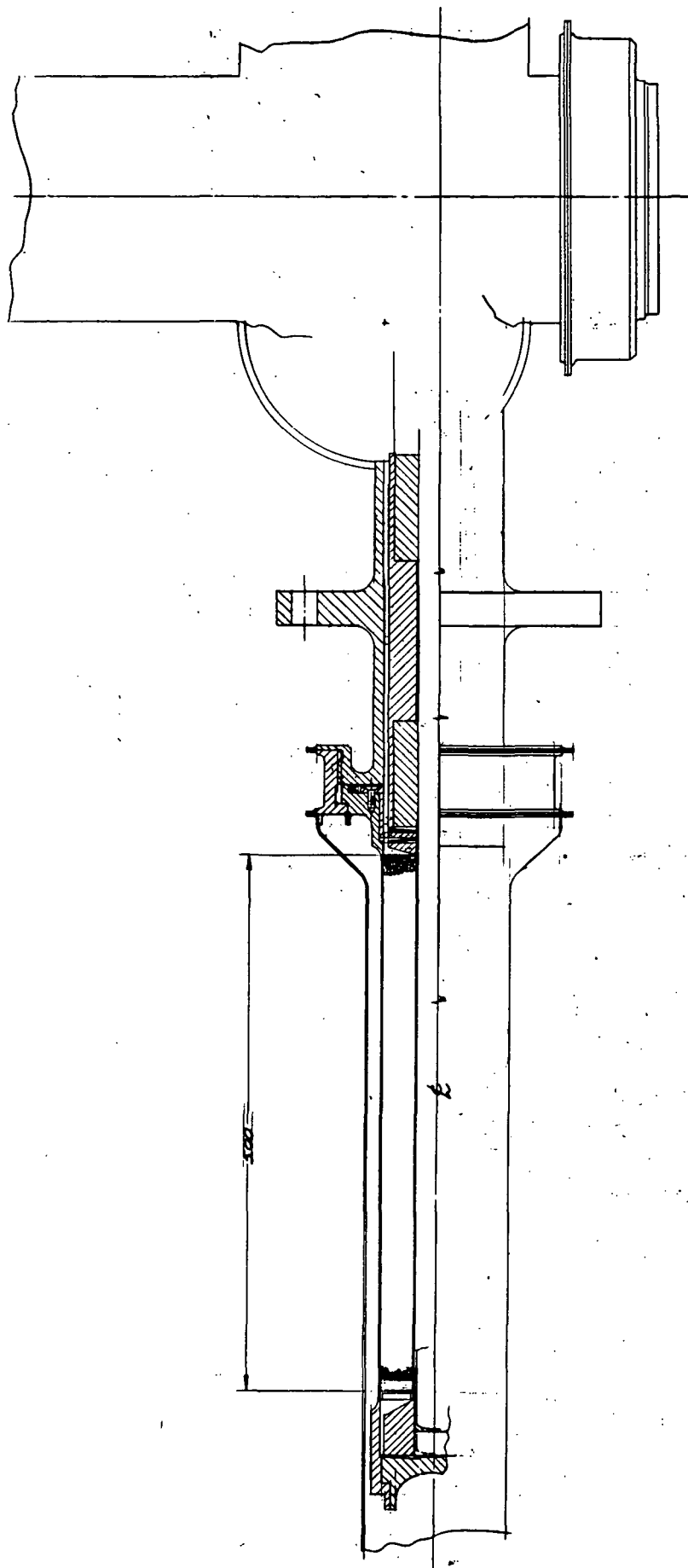


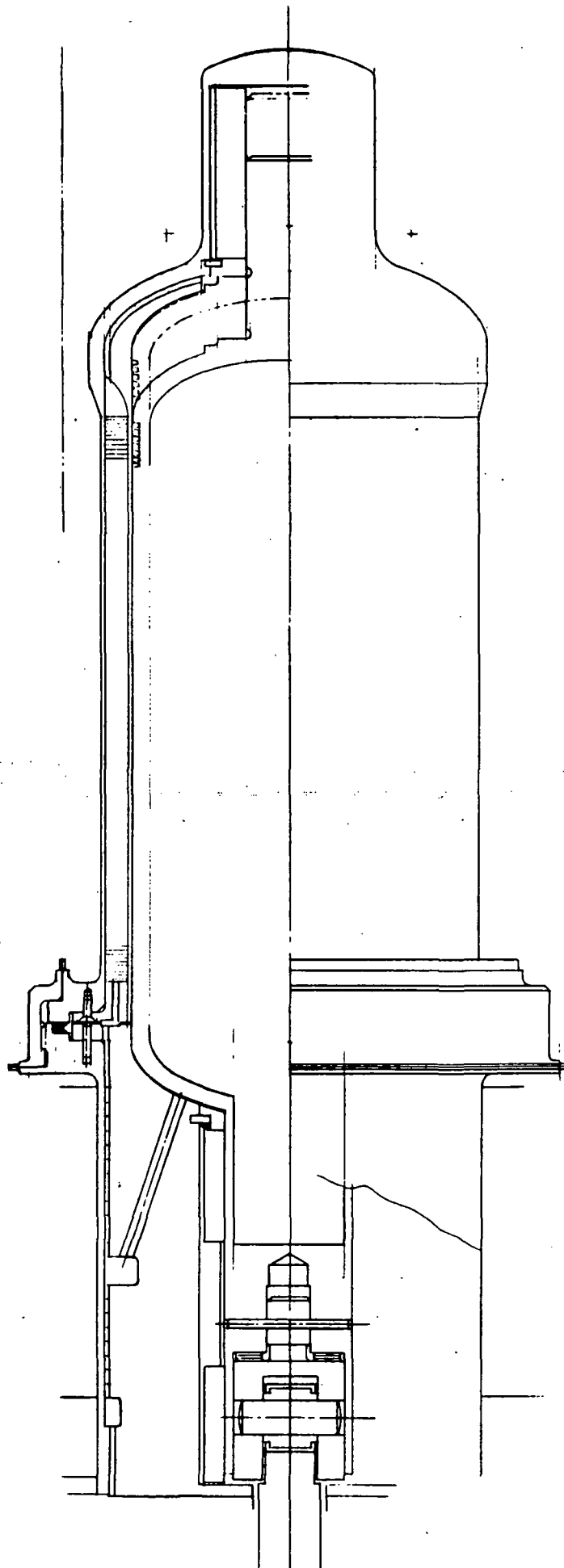
Figure 3-85. Cold End Preliminary Design



AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California

S-77595

72-8846
Page 3-162



S-77591

Figure 3-86. Hot End Preliminary Design



AIRESEARCH MANUFACTURING COMPANY
Los Angeles, California

STRESS ANALYSIS

Introduction

During the preliminary design phase (Task I), stress analyses were performed to establish and/or verify the basic dimensions of the major components. The detailed stress analyses will be performed during Task II of this program.

The preliminary stress analyses were concerned primarily with establishing the required wall thickness and stiffening ring spacing and geometry for the pressure shells, displacers, vacuum jackets, and crankcase housing. The relative motion of the cold end displacer and regenerator were also investigated for sinusoidal, random, and self-induced vibration.

Design Criteria

Although the preliminary work was performed with an operating pressure of $1.036 \times 10^7 \text{ N/m}^2$ (1500 psia), it became apparent near the end of task 1 that the heat leaks and weight penalties imposed by this pressure were too severe. A decision was therefore made to reduce the maximum operating pressure to $8.62 \times 10^6 \text{ N/m}^2$ (1250 psia). This analysis is based on the reduced operating pressure. The components subjected to working pressure were therefore analyzed for:

$$\text{Operating pressure} = 8.62 \times 10^6 \text{ N/m}^2 \text{ (1250 psia)}$$

$$\text{Proof pressure} = 1.67 \times \text{Operating Pressure} = 1.44 \times 10^7 \text{ N/m}^2 \text{ (2060 psig)}$$

$$\text{Burst pressure} = 2.25 \times \text{Operating Pressure} = 1.94 \times 10^7 \text{ N/m}^2 \text{ (2800 psig)}$$

The unit was designed to withstand proof pressure without permanent deformation and burst pressure without rupture.

All components were designed to target factors of safety of 1.15 on yield strength, 1.50 on ultimate strength, and 2.0 on elastic instability (buckling).

For components subjected to high temperatures, the allowable stress was that value at which 0.2% creep occurs in 20,000 hr. The temperature profile was 922.2°K (1200°F) at the hot end, 333.3°K (140°F) at the sump, and 65°K (-343°F) at the cold end.

For components subjected to cyclic loads, the allowable stress for steel components was based on a life requirement of 10^8 cycles.

The ultimate design load factor was 30 g applied separately in any direction. This load factor was not superimposed on the vibratory environment defined below.



The vibratory environment is as follows:

Sinusoidal Vibration - Thrust Axis ZZ

<u>Frequency, Hz</u>	<u>Level (0-to-peak)</u>	<u>Sweep Rate, Oct./min</u>
5-11	12.2 mm d.a. (0.48 in. d.a.)	2.0
11-17	±2.3 g	2.0
17-23	±6.8 g	1.5
23-200	±2.3 g	2.0

Sinusoidal Vibration - Lateral Axes X-X and Y-Y

<u>Frequency, Hz</u>	<u>Level (0-to-peak)</u>	<u>Sweep Rate, Oct./min</u>
5-9.9	19.0 mm d.a. (0.75 in. d.a.)	2.0
7.1-14	±2.0 g	2.0
14-200	±1.5 g	2.0

Random Vibration - Axes ZZ, X-X, and Y-Y

<u>Frequency, Hz</u>	<u>PSD Level, g^2/Hz</u>	<u>Accel., g-rms</u>	<u>Duration</u>
20-300	0.0029 to 0.045 increasing from 20 Hz at rate of 3 dB/octave	9.1	4 minutes each axis
300-2000	0.045	9.1	4 minutes each axis

Discussion of Results

1. Hot End Pressure Shell

With inside radius of 0.0328 m (1.290 in.), the required cylinder wall thickness based on hoop tension is 0.000741 m (0.0292 in.). The wall thickness will be increased at the ends to carry the discontinuity stresses. The thickness of the elliptical dome is 0.00165 m (0.065 in.); required to maintain the operating stress level below the creep allowable at 922.2°K (1200°F).



2. Hot End Displacer

The required cylinder wall thickness, stiffening ring spacing and geometry to resist an external pressure of $1.94 \times 10^7 \text{ N/m}^2$ (2800 psi) were computed to be:

- Cylinder wall = 0.001017 m (0.040 in.)
- Ring spacing = 0.03655 m (1.438 in.)
- Ring width = 0.00203 m (0.080 in.) (along cylinder ϕ)
- Ring thickness = 0.00254 m (0.100 in.) (radial)

3. Cold End Displacer

The configuration required to resist an external pressure of 2800 psi was:

- Cylinder wall = 0.000304 m (0.012 in.)
- Ring spacing = 0.01347 m (0.530 in.)
- Ring width = 0.000762 m (0.030 in.) (along cylinder ϕ)
- Ring thickness = 0.001142 m (0.045 in.) (radial)

4. Cold End Pressure Shell

The cylinder wall thickness was governed by the requirement that the relative motion between the displacer and the regenerator be less than the clearance provided. The estimated frequency of the displacer and the regenerator (shell) was 452 and 448 Hz, respectively. Since the frequencies are nearly equal, the motion of the displacer and shell are in phase and the relative motion is less than the clearance provided. The required cylinder wall thickness is 0.000609 m (0.024 inches) with a 0.0278 m (1.093 inch) outside diameter.

The displacement at the end of the pressure shell due to self-induced vibration was estimated at $0.508 \times 10^{-6} \text{ m}$ (20×10^{-6} inches) (double amplitude). This is an order of magnitude less than the allowable displacement of $5.08 \times 10^{-6} \text{ m}$ (200×10^{-6} inches) (double amplitude). The estimate was obtained by considering the unbalance of the displacers, connecting rods and crankshaft and driving the crankshaft at 400 rpm.



5. Crankshaft Housing

The crankshaft housing was analyzed for internal pressure. Particular attention was directed to the housing/cross tube intersection and the spherical dome/tube intersection. The required wall thickness of the housing was found to be 0.001627 m (0.064 inches) and the required wall thickness of the cross-tube was 0.00292 m (0.125 inches), and the required wall thickness of the tube at the spherical dome end was 0.00226 m (0.089 inches).

6. Crankshaft Assembly

A preliminary critical speed analysis was performed considering the effects of the displacers reciprocating linear unbalance as well as rotary unbalance. The first mode critical speed was found to be 8400 rpm; significantly higher than the 400 rpm maximum operating speed.

Analyses were also performed to verify that the crankshaft preload was sufficient to preclude separation of the stacked-parts interfaces and that the stress levels in the crankshaft and the connecting rod assemblies are at acceptable levels.

MOTOR DESIGN

The drive motor for the preliminary design VM refrigerator is a six pole squirrel cage induction motor driven by a 21 Hz, 26 volt, three phase power supply. Preliminary calculations indicate that the required power supply rating is 40 volt-amp (3 phase total) for rated load runnings, and 200 volt-amp for short term (less than a second) motor starting. The power supply should have both variable voltage and frequency output if running at different speeds will be required. The stator and the rotor laminations are made from hot rolled, low core loss, high permeability thin silicon steel to increase motor efficiency. The motor construction is mechanically rugged, with the conductor bars and end-rings integrally die case with the lamination stack. No brushes or slip rings are required.

The stator winding utilizes 493°K (220°C) wire and insulation materials to ensure long insulation life. The motor electrical and thermal design provides for low loss density to keep the maximum hot spot temperature below 493°K (220°C).

Motor performance, power factor, and efficiency is somewhat reduced by the increased air gap necessary to install an approx. 0.000254 m (0.010 inch) thick bore seal. The selection of the bore seal material is based on the combination of high electrical resistivity and high mechanical strength. Possible materials are Inconel 718, or Ti6-AL-4V Titanium alloy. The estimated motor efficiency is approximately 25%, depending on the power supply wave shape, and the estimated power factor is 40% at the rated 2.5 watts shaft output load condition. The motor is running at approximately 5% slip at the 400 rpm speed, rated load condition, and will maintain $\pm 1\%$ speed at the load for the expected temperature variations.



REFERENCES

- 3-1 Browning, C. W. and V. L. Potter, "75°K Vuilleumier Cryogenic Refrigerator, Final Report for Task II, Analytical and Test Program," AiResearch Report 72-8497, August 1972.
- 3-2 B. D. Marcus, "Theory and Design of Variable Conductance Heat Pipes Hydrodynamics and Heat Transfer," Research Report No. 1, Contract No. NAS2-5503, TRW.
- 3-3 ICICLE Feasibility Study, Final Report, NASA Contract NAS5-12039, RCA-Defense Electronic Products, Camden, New Jersey.
- 3-4 L. S. Fletcher, P. A. Smuda and D. A. Gyorog, "Thermal Contact Resistance of Selected Low-Conductance Interstitial Materials," AIAA Journal, Vol. 7, No. 7, p. 1302.
- 3-5 M. M. Yovanovich and M. Tuarze, "Experimental Evidence of Thermal Resistance at Soldered Joints", Journal of Spacecraft, Vol. 6, No. 7, p. 856, July 1969.



APPENDIX A

REGENERATOR ANALYSIS

General

The regenerator is one of the most important components of the Vuilleumier cycle refrigerator. It is a heat storage device which is used at both the hot and cold ends of the refrigerator. At the cold end, it is used to cool the gas as it flows from the sump region to the cold expansion volume of the machine. The expansion process of the gas reduces the gas temperature further in order to provide cooling at the cold temperature. After absorbing heat from the refrigeration load, the gas is then exhausted from the expansion volume through the regenerator where it regains energy previously stored, to the sump region. This reestablishes the temperature profile in the cold regenerator for the incoming gas for the next cycle.

The process in the hot regenerator is essentially the same as that at the cold end. The gas flowing from the sump region to the hot end is heated in the regenerator. As the flow reverses, an expansion process begins, which tends to cool the gas. This allows heat to be added to the gas at essentially constant temperature from the hot end heater. The gas is exhausted from the expansion volume through the regenerator where it transfers energy to the regenerator matrix and is cooled. This reestablishes the temperature profile in the hot regenerator for the incoming gas for the next cycle.

The design requirements for a good regenerator are that the regenerator packing be of a material with a very large heat capacity and that the packing have a large heat transfer coefficient and a large heat transfer area. The basic tradeoff for the design of a regenerator is between irreversible pressure drop and heat transfer potential with a minimum void volume.

The analysis of the regenerators for a Vuilleumier refrigerator cannot be based on the classical effectiveness parameters which make use of inlet and outlet temperatures. The system pressure fluctuates and a considerable amount of gas is stored in the regenerator void volume during the flow period. This results in the mass flow of gas into the regenerator not being equal to the mass flow of gas out of the regenerator at all points in time.

Computer Program

In order to analyze the regenerators properly, a computer program making use of finite difference techniques has been developed by AiResearch. The regenerator is broken into axial nodes; these separate nodes are used for both the gas and the matrix. The following basic equations were used to develop the difference equations which the computer program solves.

1. Gas Nodes

The continuity equation can be written as

$$\frac{\partial \rho}{\partial \tau} + \frac{\partial (\rho u)}{\partial x} = 0 \quad (A-1)$$



where ρ = gas density

τ = time

u = gas velocity

x = coordinate

Setting

$$G = \rho u$$

$$-\frac{\partial G}{\partial x} = \frac{\partial \rho}{\partial \tau}$$

Now writing the difference form

$$\left(\frac{G_n - G_{n+1}}{\Delta x_n} \right) = \left(\frac{\rho'_n - \rho}{\Delta \tau} \right) \quad (A-2)$$

therefore

$$G_n = G_{n+1} + \Delta x_n \left(\frac{\rho'_n - \rho}{\Delta \tau} \right) \quad (A-3)$$

The momentum equation for the gas is

$$\rho \frac{\partial u}{\partial \tau} + \rho u \frac{\partial u}{\partial x} + \frac{4f}{2D_h} (\rho u^2) + \frac{\partial P}{\partial x} g_c = 0 \quad (A-4)$$

where f = Fanning friction factor

D_h = characteristic length

P = static pressure

$$g_c = 32.2 \text{ lbf-ft/lbf-sec}^2$$

Taking the steady-state form and substituting G ,

$$g_c \frac{\partial P}{\partial x} = \frac{\partial (G^2/\rho)}{\partial x} + \left(\frac{4f}{2D_h} \right) \frac{G|G|}{\rho} \quad (A-5)$$

Now taking the difference form of this equation,

$$g_c \frac{(P_n - P_{n+1})}{(\Delta x_n)} = \frac{(G_{n+1}^2/\rho_{n+1}) - (G_n^2/\rho_n)}{(\Delta x_n)} + \frac{1}{2} \left[\left(\frac{4f_{n+1}}{2D_h} \right) \frac{G_{n+1}|G_{n+1}|}{\rho_{n+1}} + \left(\frac{4f_n}{2D_h} \right) \frac{G_n|G_n|}{\rho_n} \right] \quad (A-6)$$



Now solving for P_n

$$P_n = P_{n+1} + \frac{1}{g_c} \left(\frac{G_{n+1}}{\rho_{n+1}} \right) \left[G_{n+1} + |G_{n+1}| \left(\frac{\Delta x_{n+1}^f}{D_h} \right) \right] + \frac{1}{g_c} \left(\frac{G_n}{\rho_n} \right) \left[-G_n + |G_n| \left(\frac{\Delta x_n^f}{D_h} \right) \right] \quad (A-7)$$

The energy equation is

$$\frac{\partial(c_v \rho T)}{\partial \tau} + \frac{\partial(c_p \rho u T)}{\partial x} = \left(\frac{UA_\ell}{V_\ell} \right) (t - T) \quad (A-8)$$

where c_v = specific heat at constant volume

c_p = specific heat at constant pressure

T = gas temperature

U = overall heat transfer coefficient

A_ℓ = heat transfer surface area per unit length

V_ℓ = void volume in regenerator per unit length

t = matrix temperature

This equation can be reduced to

$$\bar{c}_p \frac{\partial T}{\partial x} + \frac{\partial(c_p GT)}{\partial x} = \left(\frac{UA}{V_\ell} \right) (t - T) \quad (A-9)$$

The overall heat transfer coefficient u is defined as

$$u = \frac{1}{\left(\frac{1}{h_c} \right) + \left(\frac{\Delta}{k} \right)} \quad (A-10)$$

$$\text{where } h_c = \left(\frac{j G c_p}{Pr^{2/3}} \right)$$

j = Colburn j -factor

Pr = Prandtl number

Δ = equivalent conduction length



Taking the difference form and solving for future gas temperature

$$T_n' = T_n + \left[\frac{(\Delta\tau_g)(G_n c_{pn} T_n - G_{n+1} c_{pn+1} T_{n+1})}{\Delta x_n \bar{c}_{vn} \bar{\rho}_n} \right] + (\Delta\tau_g) \left(\frac{UA_\ell}{\bar{c}_{vn} \bar{\rho}_n V_\ell} \right) (t_n - T_n) \quad (A-11)$$

The stability criterion for this difference equation is

$$\Delta\tau_g \leq \left[\frac{1.0}{\left(\frac{UA_\ell}{\bar{c}_{vn} \bar{\rho}_n V_\ell} \right) - \left(\frac{G_n c_{pn}}{\bar{c}_{vn} \bar{\rho}_n \Delta x_n} \right)} \right] \quad (A-12)$$

In the case that the time increment $\Delta\tau_g$ required by the stability criterion for the gas is very small compared to that of the matrix, steady-state solutions may be selected. In this case, the steady-state energy equation is

$$\frac{\partial(c_p GT)}{\partial x} = \left(\frac{UA_\ell}{V_\ell} \right) (t - T) \quad (A-13)$$

Writing the difference form

$$\frac{(G_{n+1} c_{pn+1} T_{n+1} - G_n c_{pn} T_n)}{\Delta x_n} = \left(\frac{UA_\ell}{V_\ell} \right) (t_n - T_n) \quad (A-14)$$

Solving for gas temperature

$$T_n = \frac{\left[\left(\frac{UA_\ell}{V_\ell} \right) t_n - \left(\frac{G_n c_{pn}}{\Delta x_n} \right) T_{n+1} \right]}{\left[\left(\frac{UA_\ell}{V_\ell} \right) - \left(\frac{G_n c_{pn}}{\Delta x_n} \right) \right]} \quad (A-15)$$

2. Matrix Nodes

The energy equation for the matrix is

$$\frac{\partial t}{\partial \tau} = \left(\frac{UA_\ell}{C_m M_\ell} \right) (T - t) + \frac{1}{C_m M_\ell} \frac{\partial}{\partial x} \left[k_m A_x \frac{\partial t}{\partial x} \right] \quad (A-16)$$



where t = matrix temperature

C_m = matrix specific heat

M_ℓ = matrix mass per unit length

k_m = matrix conductivity

A_x = matrix effective heat transfer area for conduction

Writing the difference equation and solving for future matrix temperature,

$$t_n' = t_n + \Delta T_m \frac{UA_\ell}{C_m M_\ell} (T_n - t_n) + \left(\frac{\Delta T_m}{C_m M_\ell \Delta x_n} \right) \left[\frac{(t_{n-1} - t_n)}{R_{n-1,n}} + \frac{(t_{n+1} - t_n)}{R_{n+1,n}} \right] \quad (A-17)$$

$$\text{where } R_{m,n} = \left[\left(\frac{\Delta x_{m/2}}{k_m A_{xm}} \right) + \left(\frac{\Delta x_{n/2}}{k_n A_{xn}} \right) + \frac{1}{(H A)_i} \right]$$

where $(H A)_i$ = interface or boundary heat transfer characteristic.

The stability criterion for the matrix time increment is

$$\Delta T_m \leq \left[\frac{1.0}{\left(\frac{UA_\ell}{C_m M_\ell} \right) + \left[\frac{1}{C_m M_\ell \Delta x_n} \right] \left[\left(\frac{1}{R_{n-1,n}} \right) + \left(\frac{1}{R_{n+1,n}} \right) \right]} \right] \quad (A-18)$$

3. Discussion

The computer program is capable of analyzing both transient performance and cyclically steady performance. The inputs required by this computer program are the physical characteristics of the regenerator, heat transfer and friction loss characteristics, initial conditions, and boundary conditions. The physical characteristics are reflected in the matrix areas, volume, length, heat capacity, thermal conductivity, and mass. The heat transfer and friction loss characteristics are read into the computer program in the form of Colburn j -factor and Fanning friction factor as a function of Reynolds number. The initial conditions must be fully described in terms of pressure and temperatures of the gas and the matrix. The boundary conditions that are required are the time dependent characteristics of pressure, mass flow rate, and gas temperature at one end of the regenerator, and the return gas temperature at the other end (basically defining the thermodynamic process at the other end).



Regenerator Characterization

In the analyses of the regenerators, friction coefficients and the Colburn j-factor have been taken from the data presented by Kays and London. The data for randomly stacked screens are shown in Figure A-1 and the data for regenerators packed with spherical shots are shown in Figure A-2. Standard screen mesh data, shown in Table A-1, was used to characterize the screen matrix. Figures A-3 and A-4 show the specific heats of Monel, stainless steel, and Inconel as a function of temperature.

1. Area to Volume Ratio

For screens, the area to volume ratio can be derived to be

$$\beta = \frac{4(1 - \epsilon)}{d} \quad (A-19)$$

where

β = area to volume ratio

ϵ = porosity

d = wire diameter

For spheres, the area to volume ratio is

$$\beta = \frac{6(1 - \epsilon)}{d} \quad (A-20)$$

where

d = sphere diameter

2. Hydraulic Diameter

Another regenerator characteristic which is required is the hydraulic diameter. Defining the hydraulic radius as being equal to the flow volume divided by the total surface area, the hydraulic radius of a screen stack is found to be

$$r_H = \frac{\epsilon d}{4(1 - \epsilon)} \quad (A-21)$$

For a sphere, the hydraulic radius is equal to

$$r_H = \frac{\epsilon d}{6(1 - \epsilon)} \quad (A-22)$$

The hydraulic diameter is equal to 4 times the hydraulic radius.



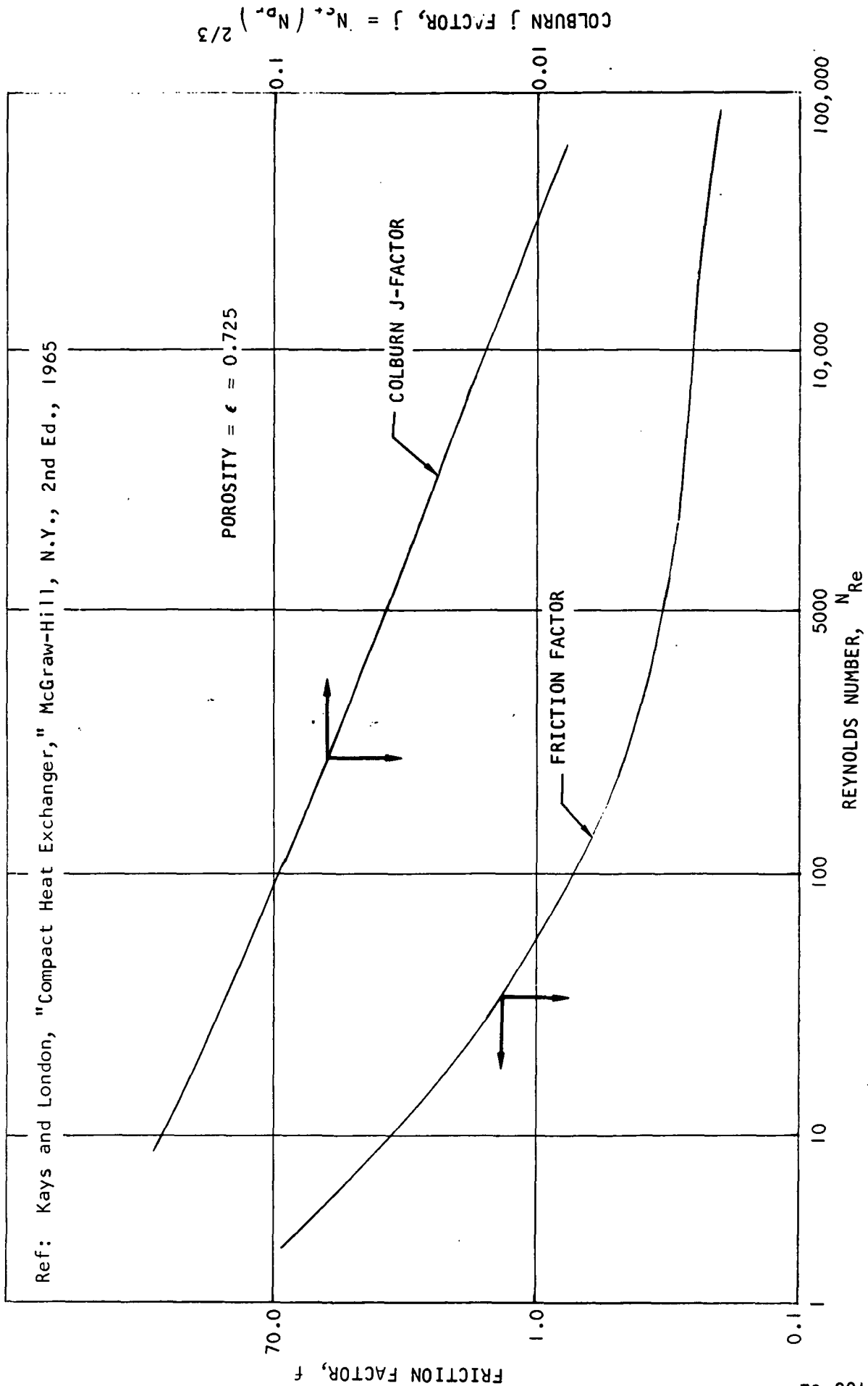
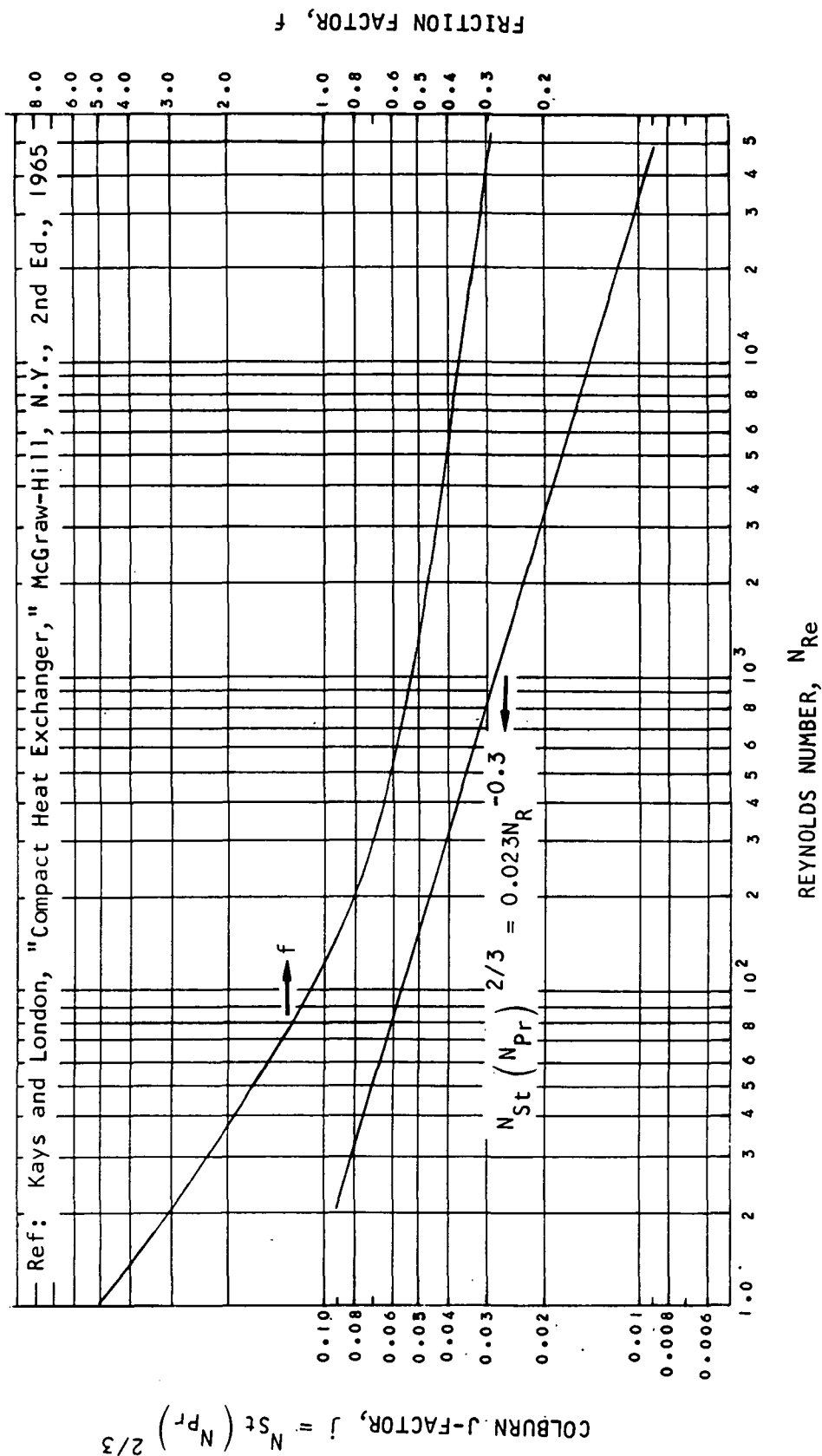


Figure A-1. Heat Transfer and Pressure Drop Data for Stacked Screens

S-58836



S-58837

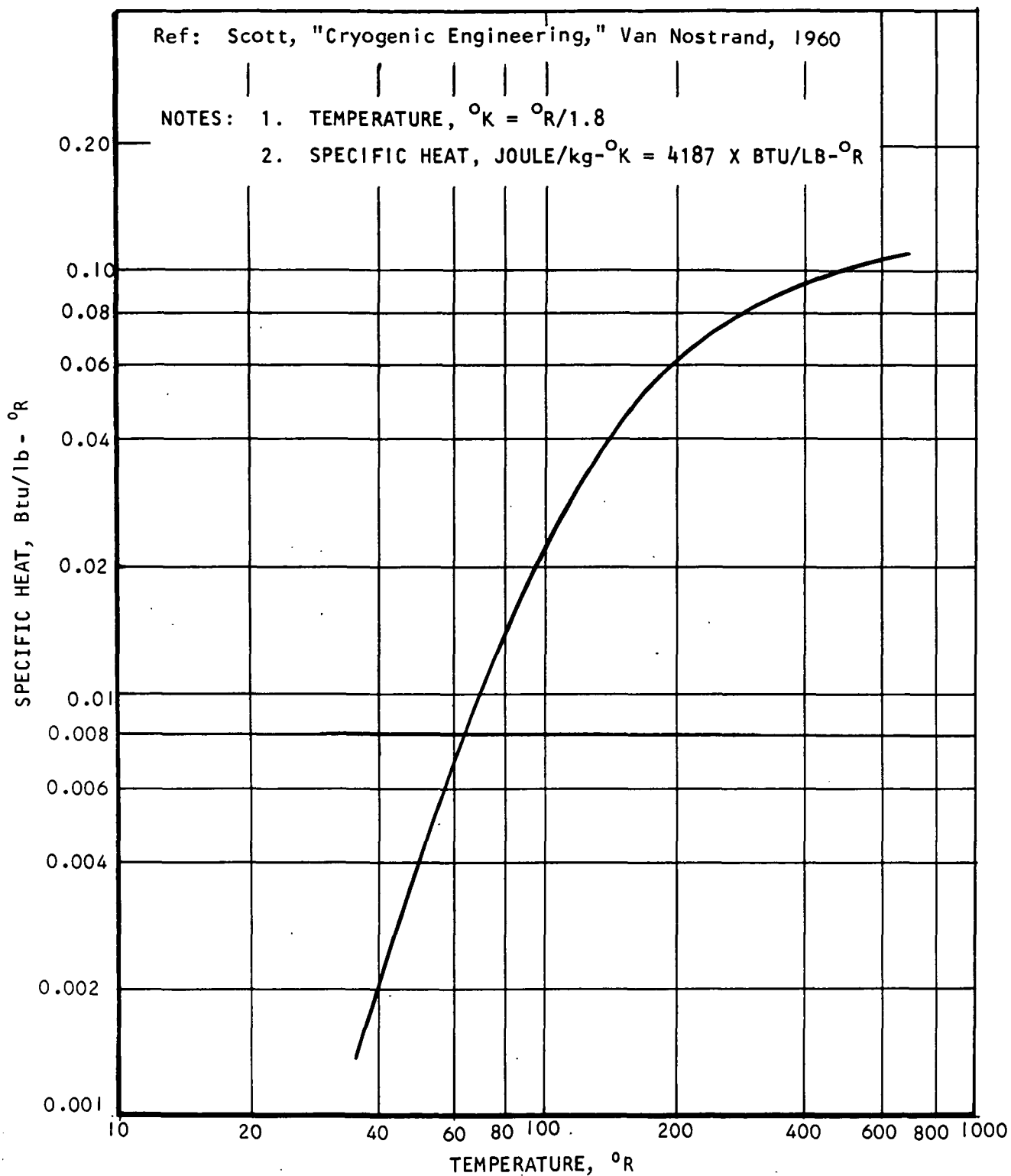
Figure A-2. Gas Flow through an Infinite Randomly Stacked Sphere Matrix

TABLE A-1
U.S. SIEVE SERIES AND
TYLER EQUIVALENTS
A.S.T.M. - E-11-61

Sieve Designation		Sieve Opening		Nominal Wire Diameter		
Standard	Alternate, No.	mm	Inches, approx. equiv.	mm	Inches, approx. equiv.	Tyler Equiv. Designation, mesh
1.41 mm*	14	1.41	0.0555	0.725	0.0285	12
1.19 mm	16	1.19	0.0469	0.650	0.0256	14
1.00 mm*	18	1.00	0.0394	0.580	0.0228	16
841 micron	20	0.841	0.0331	0.510	0.0201	20
707 micron*	25	0.707	0.0278	0.450	0.0177	24
595 micron	30	0.595	0.0234	0.390	0.0154	28
500 micron*	35	0.500	0.0197	0.340	0.0134	32
420 micron	40	0.420	0.0165	0.290	0.0113	35
354 micron*	45	0.354	0.0129	0.247	0.0097	42
297 micron	50	0.297	0.0117	0.215	0.0085	48
250 micron*	60	0.250	0.0098	0.180	0.0071	60
210 micron	70	0.210	0.0083	0.152	0.0060	65
177 micron*	80	0.177	0.0070	0.131	0.0052	80
149 micron	100	0.149	0.0059	0.110	0.0043	100
125 micron*	120	0.125	0.0049	0.091	0.0036	115
105 micron	140	0.105	0.0041	0.076	0.0030	150
88 micron*	170	0.088	0.0035	0.064	0.0025	170
74 micron	200	0.074	0.0029	0.053	0.0021	200
63 micron*	230	0.063	0.0024	0.044	0.0017	250
53 micron	270	0.053	0.0021	0.037	0.0015	270
44 micron*	325	0.044	0.0017	0.030	0.0012	325
37 micron	400	0.037	0.0015	0.025	0.0010	400

*These sieves correspond to those proposed as an international (ISO) standard.
It is recommended that wherever possible these sieves be included in all sieve analysis data or reports intended for international publication.





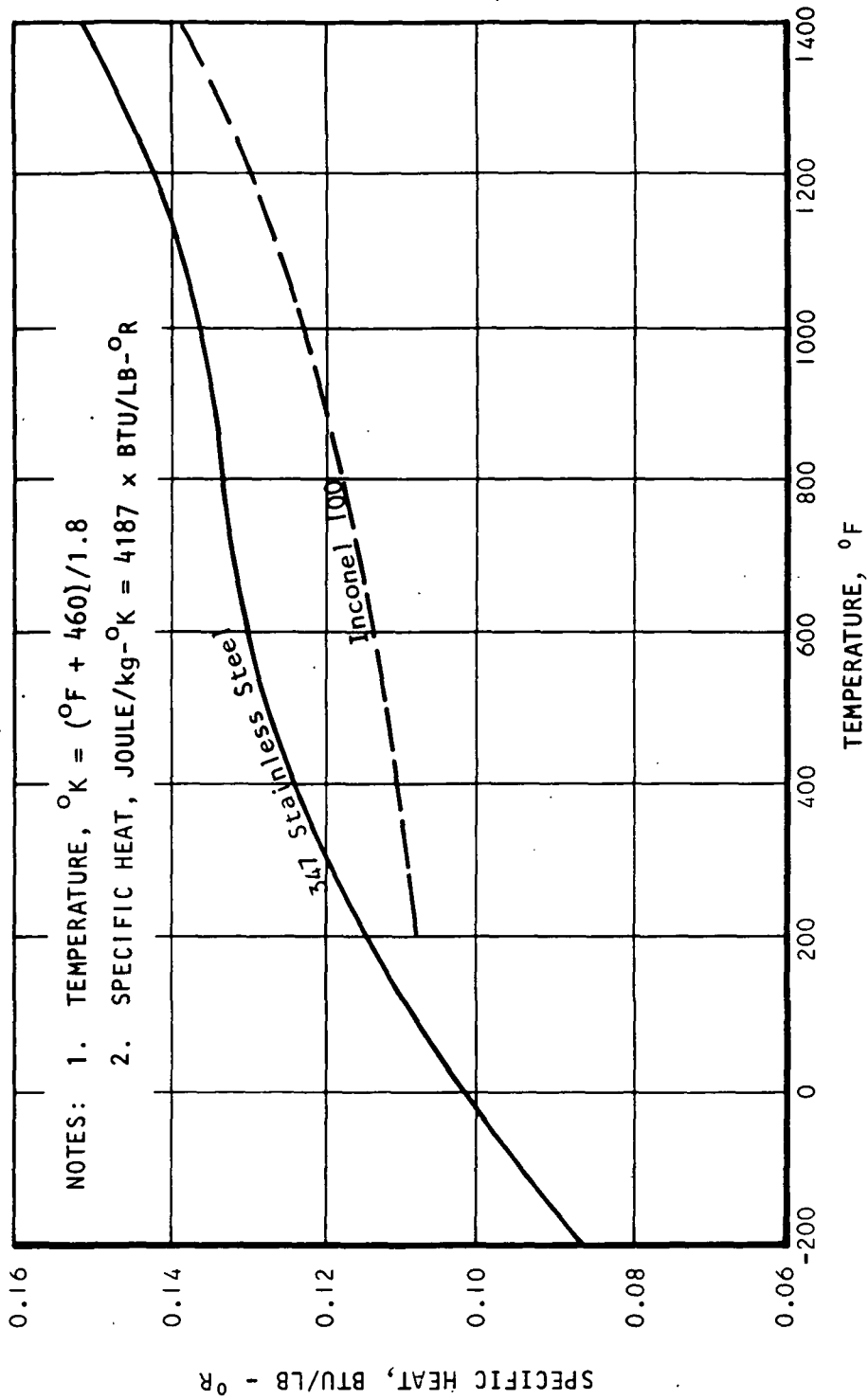
S-58838

Figure A-3. Monel Specific Heat





Ref: AiResearch Material Design Data
347 STAINLESS STEEL: ASTM STP NO. 227
INCONEL 100: G.E. HANDBOOK, SEC. 2



S-62253

Figure A-4. Stainless Steel and Inconel Specific Heats

APPENDIX B

FINAL CLOSURE SEALING WELDING TEST PLAN

INTRODUCTION

The design and function of a closure technique for sealing all entrance ports is critical to efficient operation of the fractional watt VM. The concept developed under this contract involves a metallic temporary sealing mechanism coupled with a permanent weld seal following all fabrication operations. An illustration of a candidate design is shown in Figure B-1. The attributes of this device may be summarized as follows:

- (a) Preliminary sealing is accomplished by a metallic sealing device (a "k" seal or metal "O" ring) thus allowing repeated sealing without altering the components
- (b) All mechanical and pneumatic loads are carried by the threaded retainer
- (c) Following final entrance into the VM, the two pieces being sealed are welded together at the ring. The orientation and position of the two pieces is maintained by a typical weld restraining tool shown in Figure B-2. This method allows the complete freedom needed during a development program but also provides, at minimum weight, the reliable leak-tight sealing required of a 2 year flight program

These concepts must be confirmed by a comprehensive test program to establish complete compliance with the requirements of the overall project. This may be accomplished by subjecting simulated closure mechanisms to the conditions and requirements anticipated for the flight model VM.

TEST SAMPLES

Figure B-2 illustrates a typical tool necessary to restrain the two pressure vessel halves while the threaded ring is removed and replaced prior to the final closure weld. Figures B-3 and B-4 depict the hot-end closure assembly and the cold-end closure assembly respectively.

In each instance, the exact configuration currently envisioned in the actual VM has been incorporated in the test models. In this way, the levels of stress that will be encountered in this test program will duplicate those that will be found in the actual engine.

TEST PROCEDURE

The test procedure for evaluating the acceptability of the closure concept will be the same for the hot-end and cold-end configurations illustrated in Figures B-3 and B-4. The test pieces will be fabricated



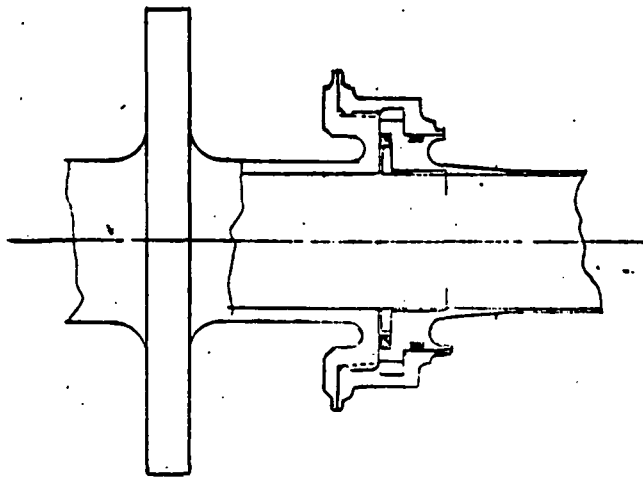
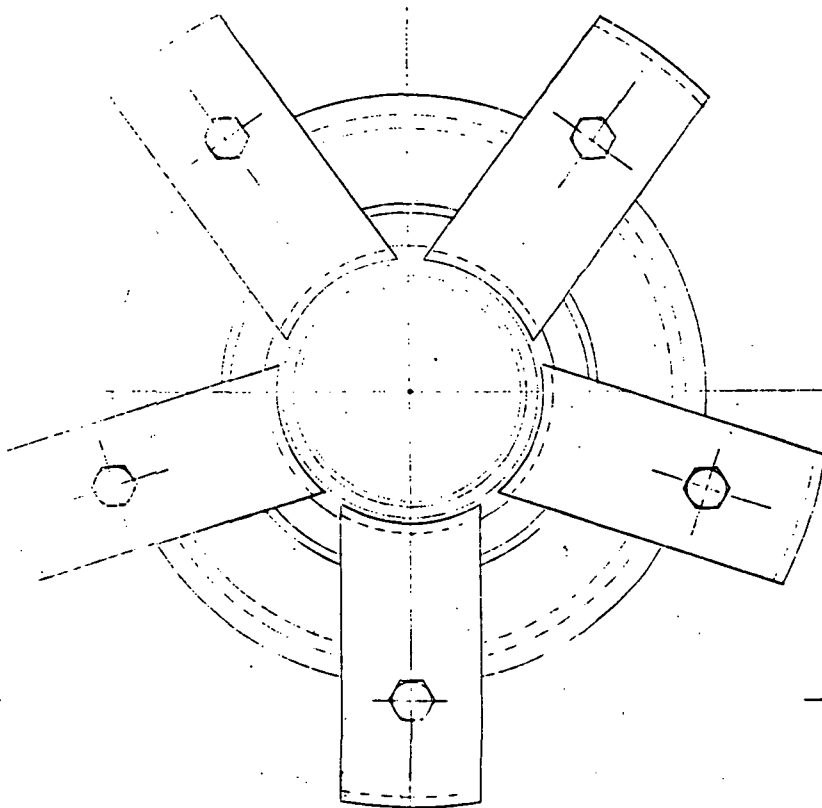
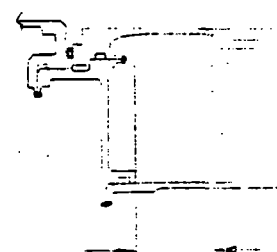
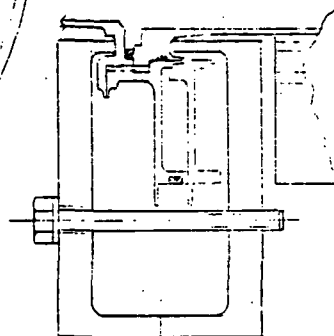


Figure B-1. Candidate Closure Design

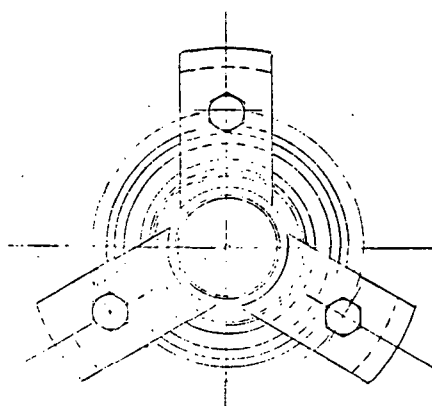




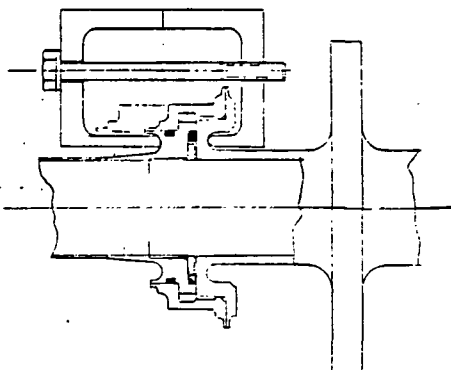
HOT DISPLACER HOUSING
ASSEMBLY TOOL

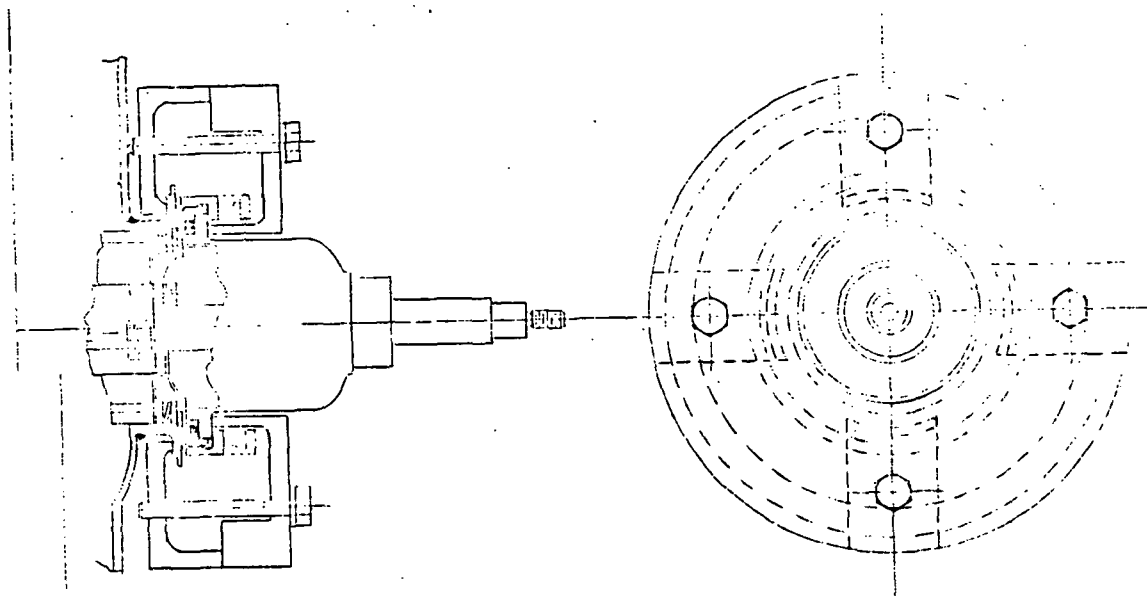


REMOVABLE
SEGMENT OF
INSULATION

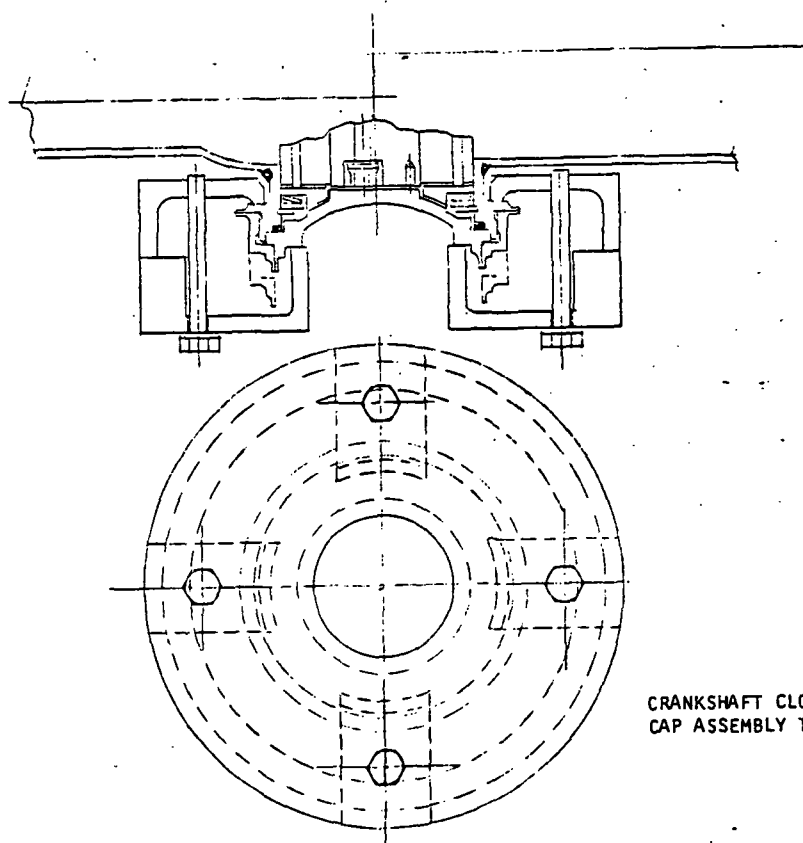


COLD DISPLACER HOUSING
ASSEMBLY TOOL





BREAK-IN DRIVE ASSEMBLY TOOL



CRANKSHAFT CLOSURE
CAP ASSEMBLY TOOL

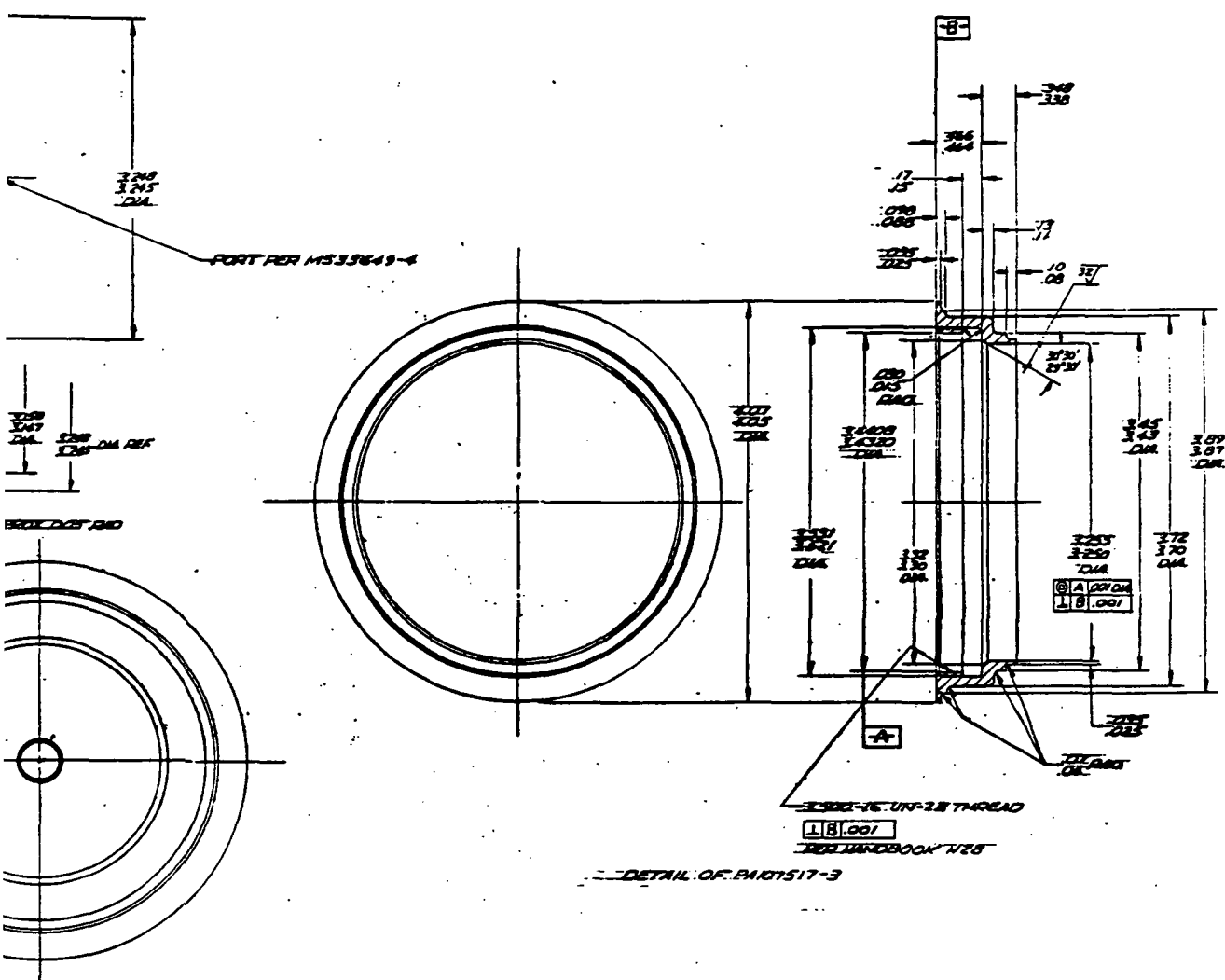
S-77554

Figure B-2. Restraining Tools - Typical

Reproduced from
best available copy.



B-3-a B

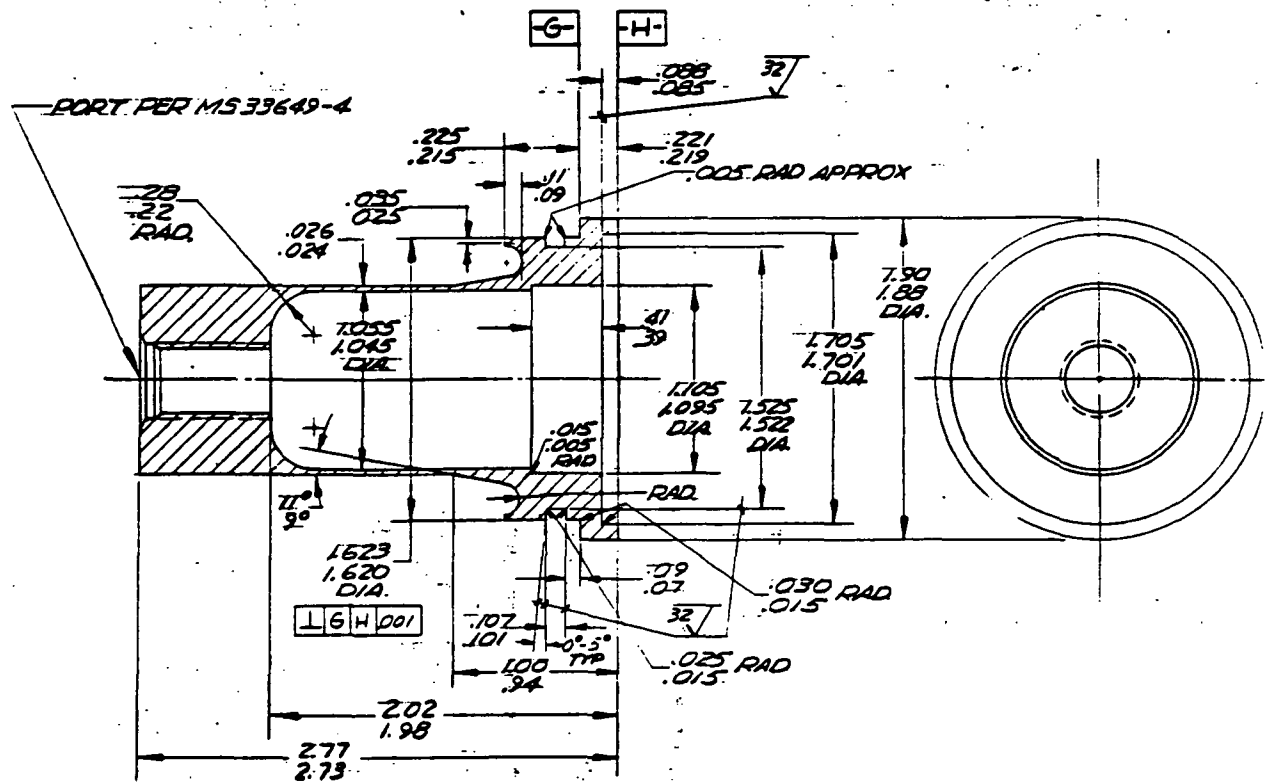
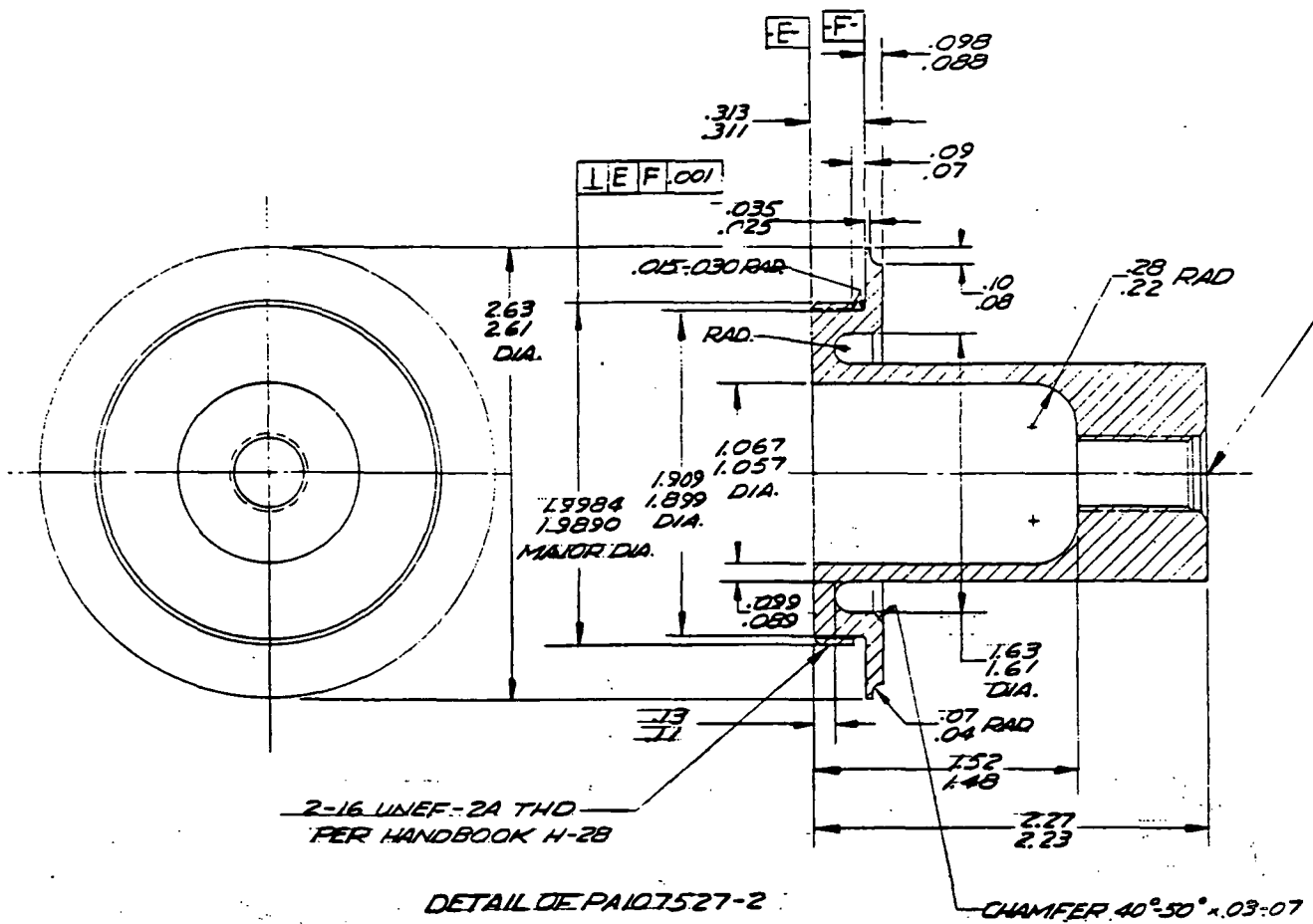


S-77594

Figure B-3. Hot-End Closure Assembly

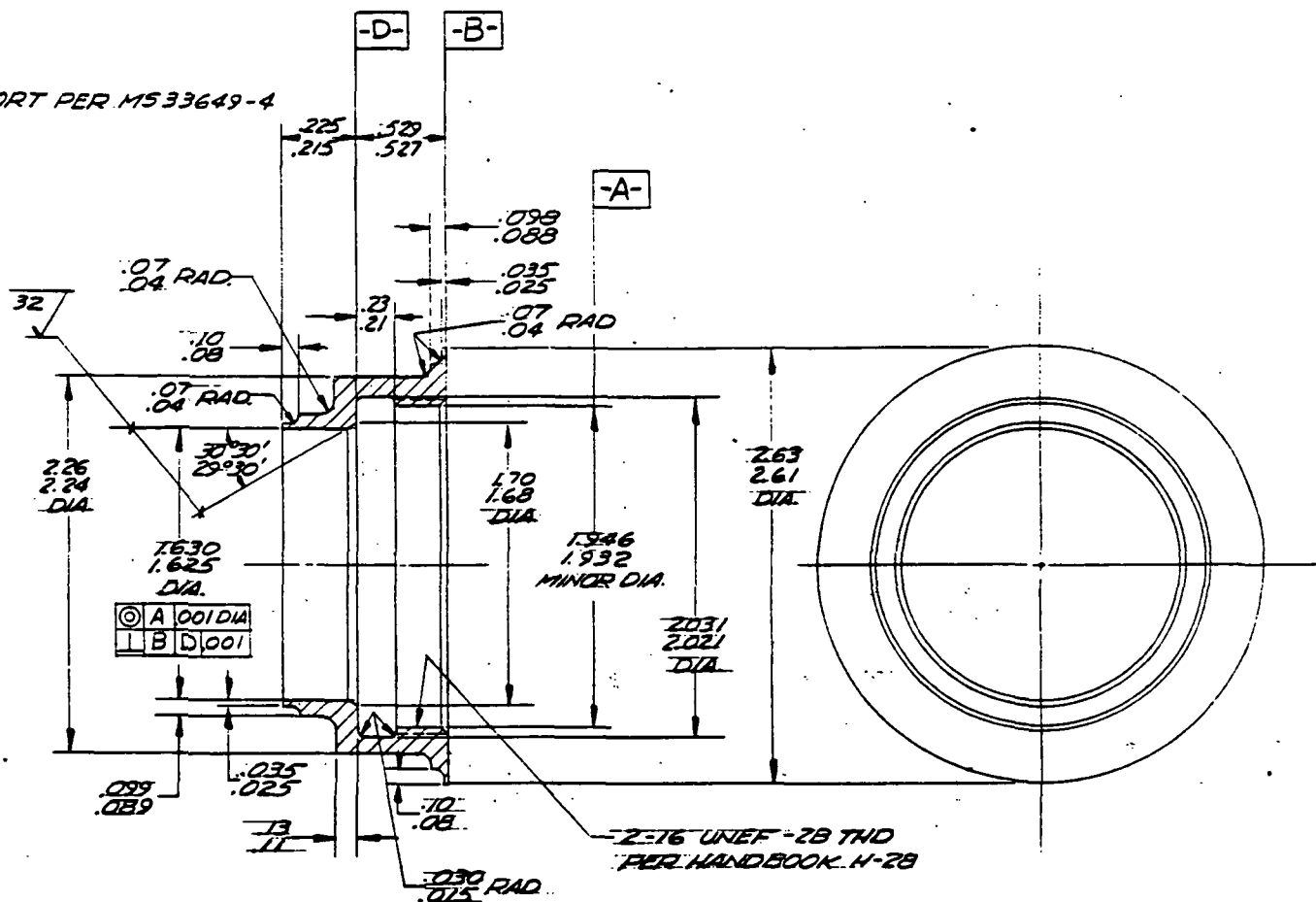
B-4-a

8



1. ALL MACHINED SURFAC

PORT PER MS33649-4



DETAIL OF PA107527-1

S-77593

S 1257

Figure B-4. Cold-End Closure Assembly

B-5-a

in accordance with applicable engineering drawings and the metallic seals obtained from a commercial source.

The test pair will be assembled, incorporating the metallic seals, and after assembly the test specimen will be mechanically loaded with the appropriate assembly tool. Once the restraining ring is torqued into place, the assembly tool will be removed and the sealed assembly subjected to a helium mass spectrometer leakage test. Once it has been established that the leakage rate is 1×10^{-7} standard cubic centimeters per second helium or less, with $1.034 \times 10^7 \text{ N/m}^2$ (1500 psig) applied to the specimen internally, the sealing pair will be taken apart and reclosed. The test setup diagram shown in Figure B-5 illustrates the leak detection method. The operations described above will be repeated four times to verify that a laboratory-grade seal is established that will allow operation of the VM prior to final weld closure.

After the four closure operations are completed, the test specimen will be subjected to the following operations:

1. Install the appropriate closure tool to restrain the test specimen in its original position
2. Remove the threaded restraining ring
3. Pierce the metallic sealing member to destroy the seal
4. Reinstall the threaded restraining ring and torque to required level
5. Perform helium mass spectrometer leak tests to determine destruction of seal
6. Perform closure welds
7. Perform helium mass spectrometer tests of closure welds--leakage not to exceed 1×10^{-10} scc/sec helium
8. Grind closure welds to allow removal of the threaded restraining rings and repeat items 6 and 7 as a weld or component repair operation

Upon completion of this test sequence, all data gathered will be cataloged and reported in the applicable monthly progress report to allow evaluation by the cognizant personnel.



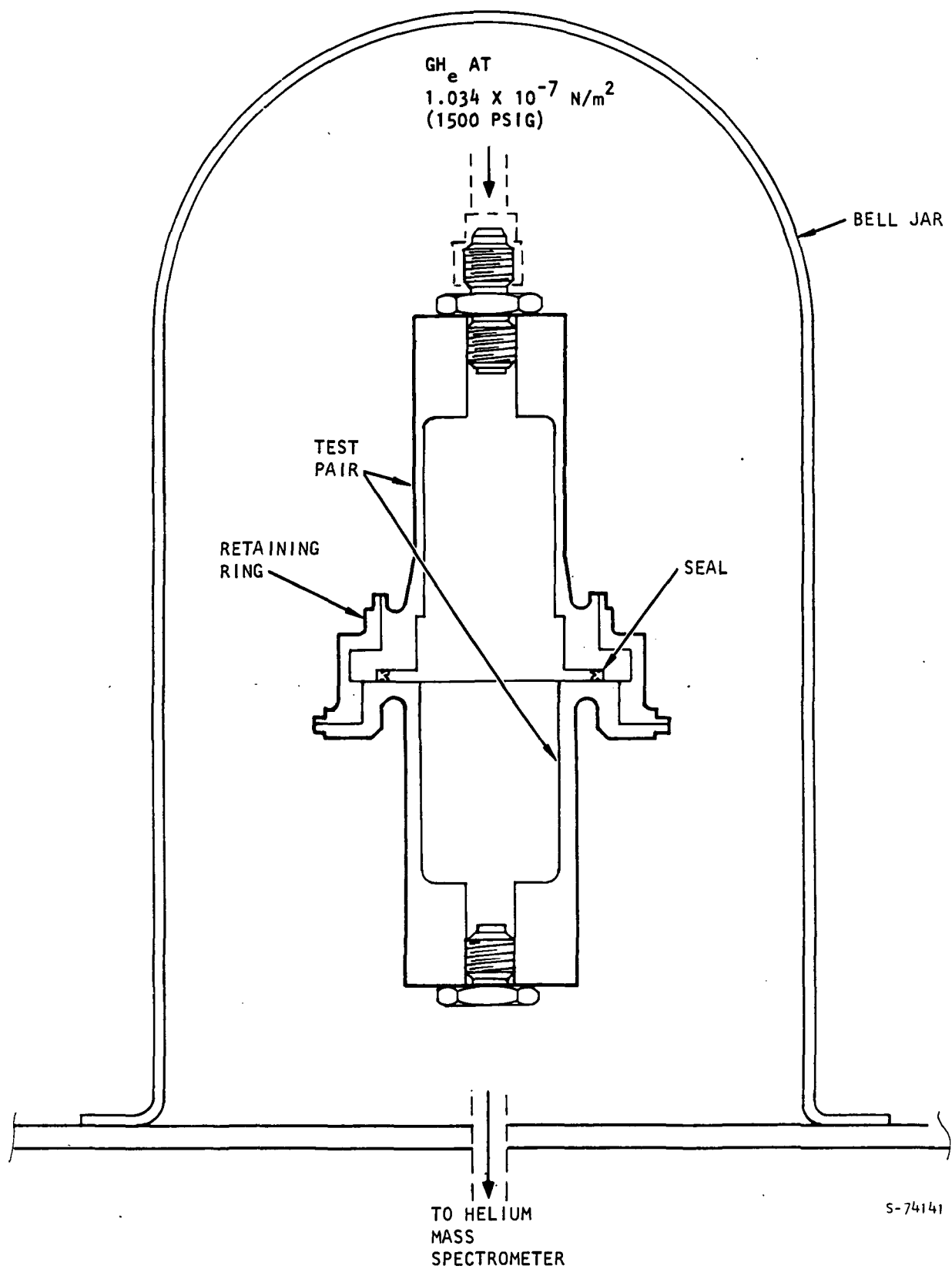


Figure B-5. System Test Setup Diagram

